STEAM TURBINES

PRACTICE AND THEORY

BY

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PREFACE

This book had its beginning in the editorial office of *Machinery*, New York. For nine years previous to July, 1906, the author was editor-in-chief of *Machinery*, and it was during the latter half of this period that the steam turbine became a commercial success in America and finally reached its dominant position in the field of electric generating.

Inasmuch as a live technical journal is constantly in touch with events in its field, the author had an exceptional opportunity to collect notes and data on the steam turbine and allied subjects during the important formative period of the turbine, when it was more discussed, probably, than any engineering subject. Advantage was taken of this opportunity to begin what has grown to be a large collection of such notes and data, which have served as the foundation for the present volume.

The author is responsible for various articles and paragraphs which have appeared in *Machinery* upon the subject of the turbine, some of which are here reproduced in more or less modified form. Most of the matter in these pages, however, was written especially for them and has not been used before.

In brief explanation of the contents, it may be said that the first chapter is a condensed treatise upon the fundamental principles of the steam turbine. The second chapter traces its early development and shows the "state of the art" at the time when the turbine became a commercial success. It is believed that this review will not only save considerable research, but in connection with the first chapter will serve to ground the beginner in first principles; for there is no way of accomplishing this so effectively as by a study of what others have done.

Much attention has been devoted to the results of tests upon the flow of steam, upon the action of steam on vanes and upon the economic performance of turbines. For comparison with the latter there is also a review of tests upon reciprocating engines, and enough data are included to enable an intelligent comparison to

be made under different running conditions, with due allowance for the efficiency of generators and engines.

The mathematical treatment has been limited mainly to a discussion of the adiabatic flow of steam and to the principles of turbine vanes, which together are the basis of all turbine calculations. Preliminary to the calculations upon the flow of steam is a chapter for reference, upon the properties of steam, which includes, also, an explanation of the temperature-entropy diagram, and data upon the specific heat of superheated steam—subjects constantly recurring in modern writings upon heat and steam. The attempt has been to simplify the mathematical treatment as much as possible, and illustrative examples have been worked out wherever it was thought they would be helpful.

The commercial and operative sides of the subject have received due attention, since no broad grasp of any important engineering subject can be had by viewing it solely from the standpoint of the technicist. Under this heading may also be classed the treatment that is given of high-vacuum condensing systems, which it is generally admitted cause engineers more trouble than the turbines themselves.

Other reference to the contents seems unnecessary except to say that the descriptive part of the text was intended to be comprehensive, but free from padding. The author wishes to express his appreciation for assistance rendered and information given by many engineers and friends; to the publication departments of the several turbine manufacturing companies for material so courteously supplied; and to technical periodicals from which information has been drawn and to which credit has been given at the proper places throughout the text. The author is especially indebted to The Industrial Press, publishers of *Machinery*, for permission to use matter and engravings that have appeared in the columns of that journal.

L. G. F.

Brattleboro, Vt., January, 1907.

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STEAM TURBINES

CHAPTER I

STEAM TURBINE PRINCIPLES.

The steam turbine, like the water turbine, is based on the principle that when a fluid is in motion its energy will be converted into mechanical work, if the fluid impinges on moving vanes which change its direction of flow and reduce its velocity. It differs from the water turbine in important particulars, however, due to the facts that water and steam have very different properties and that the steam turbine, like the steam engine, is a heat motor and must utilize the heat energy of steam.

The Principle of the Water Turbine is illustrated in Fig. 1, which shows the effect of a curved vane upon a stream of water.

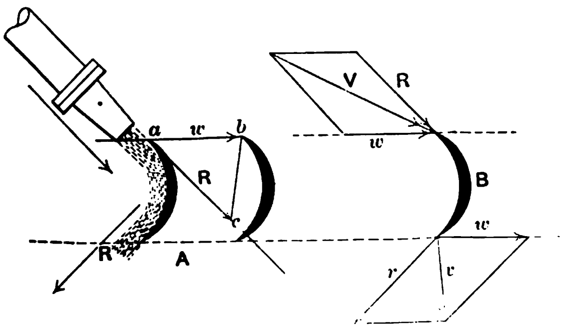


Fig. 1. Effects of Stationary and Moving Vanes upon a Stream.

The lines w, R, V, etc., represent velocities and also show direction of motion. At A the vane is supposed to be stationary and the stream glides upon it tangentially, without shock, at a

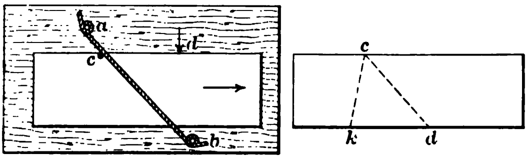
velocity R, and leaves it tangentially at the same velocity. The only effect of the vane is to alter the direction of flow.

If the vane now be given a velocity w in the direction shown, a particle of water starting at a will reach point c by the time the tip of the vane has traveled from a to b. The direction and velocity of the stream relative to the vane will then be represented by the line bc, and it is evident that the water will meet the vane with considerable shock and in this instance will fail to touch it anywhere except at the tip.

To bring about tangential action of the jet while the vane is moving, which is essential to smooth and economical running,* either the nozzle must be given a motion w, or else the direction and velocity of the stream must be changed to V, the resultant of R and w. (Shown at B.) The motion of the water relative to the moving vane will then be the same as its motion relative to the stationary vane before the change was made. Similarly, in leaving the vane the motion r, relative to the vane is tangential and equal to R. But as the stream and vane each has the velocity w at the exit, the absolute or real velocity of the stream will be the resultant of r and w, or v.† The difference between V and v represents that part of the stream velocity which, neglecting losses, has been transformed into wheel velocity w. We thus have:

*This is the theoretical statement of the conditions.

†The subject of relative motion can be made clear by the simple device herewith. Two pins, a and b, are driven in a board and a cord is stretched between them. Place an oblong piece of paper under the cord in the position shown and, holding it sta-



tionary with one hand, draw a diagonal line across the paper by following the cord with the point of a pencil. Remove the paper and c d (at the right) will be the line drawn. Again place the paper in its former position and run the pencil along the cord, at the same time pulling the paper to the right just fast enough so that when the pencil reaches the opposite edge of the paper point c will come opposite arrow d marked on the board. The line traced this time will be c k. In both cases the pencil followed the same course, *relative to the board*, but its motion *relative to the paper* was not the same when the paper moved. In like manner the motion, *relative to the vane*, of the jet of water in Fig. 1 is different, when the vane moves, from what it is when the vane is stationary.

Energy given to the moving vane by each pound of water =

$$\frac{V^2 - v^2}{2g}$$

In this case, therefore, the effect of the vane is not only to change the direction of the stream, but to reduce its velocity, by which means its kinetic energy is changed into mechanical work at the turbine wheel.

Impulse and Reaction.

Impulse and Reaction of a Jet.—The dynamic pressure upon the vane of a turbine, which causes the rotation of the wheel, is the result of the impulse and reaction of the impinging jet. According to older writers in mechanics, an impulse is a force acting in a forward direction and a reaction is the equal and opposite force—a force acting in a backward direction relative to the impulse. These are the commonly accepted meanings, although strictly an impulse is not a force in the sense of being a push or a pull, but is a term used to express the same meaning as momentum, when a force acts for a very short time, as when one body gives another a sharp blow. A reaction is properly defined as a force.

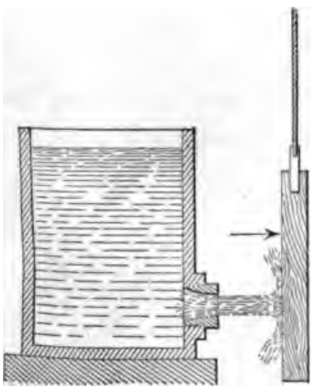


Fig. 2. Impulse.

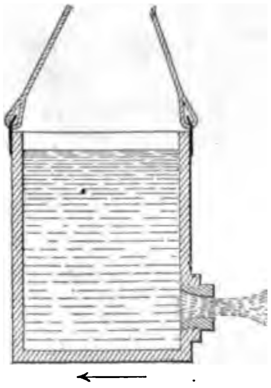


Fig. 3. Reaction.

In Fig. 2 is a tank from which water issues through a nozzle and impinges against a flat surface capable of moving horizontally —in this case the face of a plank suspended so that it is free to

swing. When the water strikes the plank the latter will swing to the right under the pressure due to the impulse of the jet. As the jet leaves the nozzle, however, it exerts a reaction against the tank, which is equal and opposite to the force due to the impulse of the jet. This may be illustrated by suspending the tank, as in Fig. 3, so that it is free to swing, when it will move to the left, owing to the reaction of the jet.

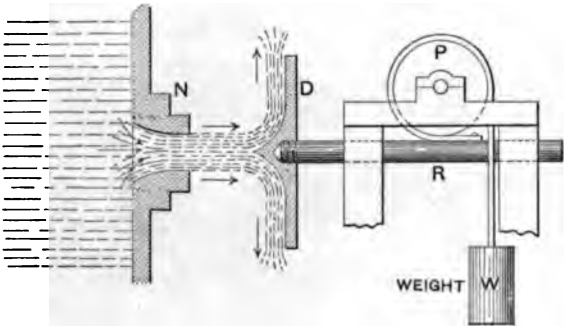


Fig. 4. Measuring Pressure due to Impulse.

The explanation of the reaction is that there is no pressure on that part of the tank where the nozzle is situated and the unbalanced pressure on that part directly opposite to the nozzle will therefore tend to move the tank in a direction opposite to that in which the water is escaping. The faster the water escapes, the greater the pressure required in the tank to give the water its velocity and hence the greater the unbalanced pressure which we call the reaction.*

Impulse and Reaction Upon Curved Vanes.—In Fig. 2 the velocity of the jet is suddenly checked when it strikes the plank, and there is more or less commotion of the particles of the fluid, causing loss of energy. This is always the case when there is impact of the particles against a surface and is to be avoided by using a curved surface so placed as to change the direction of flow gradually without shock or jar, as previously explained.

*The reaction is not determined solely by the pressure in the tank. Since action and reaction are always equal, the reaction upon the tank must be that due to the momentum of the jet. The velocity and weight of fluid discharged, however, and hence the momentum, depend in part upon the shape of the nozzle, and its *effective* area, which is never exactly equal to the measured area, owing to the coefficient of contraction of the jet.

Such an arrangement is shown in Fig. 4, where a nozzle N is discharging against a plate D attached to the rod R. The rod is supported by guides and together with the plate can move longitudinally. The pressure of the jet against the plate is balanced by the weight W supported by a cord passing around the pulley P and attached to the rod R. The plate D is so shaped at the center as to gradually guide the particles of water through an angle of 90 degrees, thus avoiding the impact present in Fig. 2, and the particles leave the plate in a direction parallel with its face. It is evident that the pressure against the plate is due solely to the impulse of the jet and that the reaction of the water in leaving the plate has no tendency to move the plate longitudinally.

In Fig. 5 the case is different. Here the water strikes a curved surface and is turned back upon itself through an angle of 180 degrees. This surface is therefore acted upon by two forces, both

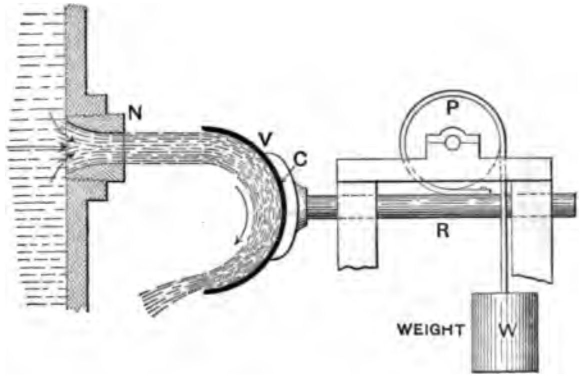


Fig. 5. Measuring the Combined Effect of Impulse and Reaction.

tending to move it to the right. The first is that due to the impulse of the jet, just as in Fig. 4, which acts until the central point C of the curved surface is reached; and the second is the reaction of the jet, which begins where the jet starts to flow backward and continues up to the edge where it discharges. These forces would be equal if there were no frictional or other losses, and it would require a weight, W, just twice as heavy as the weight in the first example to balance the end thrust.

Essential Features of the Turbine.

How Steam and Water Turbines Differ.—In a steam turbine the energy of the steam, due to its motion, is converted into mechanical work at the turbine wheel by the use of curved vanes, as outlined for water turbines. There are two important distinctions between steam and water turbines, however, which it will be advisable to refer to here. These are:

First, provision must be made in the steam turbine for converting the heat energy of the steam into kinetic energy, or the energy of motion. To accomplish this the passages and nozzles of the steam turbine must be designed to control the expansion of the steam in a way to augment its velocity. In the De Laval turbine the expansion is effected in an "expanding nozzle" which directs the steam jet against the blades of the wheel. The walls of the nozzle diverge in the direction of flow of the steam so that its outlet area is larger than its inlet area, whereas, in a water nozzle the outlet area would be smaller than the inlet area.

Second, the steam turbine must be adapted to the high velocities of steam, which have no parallel in hydraulic work, although velocities of over 300 feet a second are met with in water power plants on the Pacific slope. The enormously high velocity with which steam flows from an orifice often exceeds the speed of a rifle bullet. The new Springfield rifle adopted by the United States Army gives an initial velocity to its bullet of 2,300 feet per second, or over 26 miles a minute, and this is almost exactly the velocity with which steam at 50 pounds gauge pressure would issue from a nozzle of the best shape when discharging into the atmosphere. In single-wheel impulse turbines, operating under higher pressures and with a condenser, velocities of 3,000 to 4,000 feet per second, or even more, are attained. The problem of dealing with a fluid capable of attaining such incomprehensible velocities calls for serious consideration. By some means or other the rotating elements must be kept within the speed of safety to guard against rupture of material, cutting of bearings, etc., and at the point where power is to be used the speed must be within the practical limits of convenience.

The Successful Steam Turbine.—It will be evident from the

foregoing that the successful steam turbine must accomplish the three following results:

First, as much of the heat energy of the steam as possible must be converted into kinetic energy.

Second, the wheel must be capable of utilizing the kinetic energy of the steam in an efficient manner.

Third, the apparatus must run at a moderate speed at the point where it delivers its power, and all parts must be kept within the speed of safety.

The Steam Turbine Compared with the Steam Engine.—The fundamental principle in any economical steam motor, whether turbine or piston engine, is that the expansive force of the steam must be utilized. The direct-acting steam pump, although serving a very useful purpose, is one of the most wasteful steam users in existence. Steam is supplied to the cylinder at boiler pressure throughout the whole stroke. It forces the piston ahead because of the static pressure back of it in the boiler, in the same manner that water would do if the steam cylinder of the pump were connected to a city water main. Such expansion as occurs takes place in the boiler at constant temperature.

Steam, however, is capable of much better use than this, because it is supplied with a store of heat energy which has the power to make the steam in the cylinder expand and push the piston forward, after all communication with the boiler has been cut off. In an engine working expansively steam is admitted at boiler pressure until the point of cut-off is reached. Up to this event the action is the same as in the steam pump, but during the rest of the stroke the piston is pushed ahead as a result of the heat energy of the steam encased in the cylinder.

In the steam turbine the process of the expansion engine is duplicated, except that the flow of the steam is continuous instead of intermittent. It was explained that steam is first forced into the engine cylinder by the pressure in the boiler and then is allowed to expand in virtue of its own internal heat energy. In the turbine the steam is continuously pushed into the nozzle by the higher pressure at the inlet, and during the passage through the nozzle it expands continuously because of its internal energy. Each particle, as it expands, pushes the particles ahead of it for-

ward at a faster rate and so increases the velocity of flow. Although the turbine and piston engines are different in outward form, they are equivalent in the thermodynamic action.

The difference in the form of the turbine and engine is due to the fact that the turbine is designed to operate by changing the motion of flowing steam, on the principle of the water turbine, while the engine is designed to operate by the direct pressure of the steam. The turbine is a velocity motor and the steam engine a pressure motor.*

Steam Nozzles.

The Study of Steam Nozzles of Great Importance.—Since a turbine operates by changing the motion of flowing steam, attention must be given to the proportions of the passages through which the steam flows. In the De Laval and some other turbines, the steam flows through nozzles which direct it against the blades of the rotating wheel. In other machines it flows through passages between guide vanes which form what is virtually a group of nozzles. In still others of the Parsons type both the stationary guide vanes and the blades of the wheel have the same functions that a collection of nozzles placed side by side would have. Whatever the arrangement of the passages of a turbine, through which the steam passes, they may be regarded as steam nozzles, provided the steam fills them completely, leaving no air spaces, just as water fills completely all the space in a water nozzle used in connection with a hose pipe. In order to understand the action of steam flowing through the passages of a turbine, it is necessary to study its action in flowing through nozzles of different shapes, and the principles discovered may then be applied in proportioning the ducts or passages of the turbine.

*It must not be imagined that the driving force in a turbine is the statical steam pressure. All turbines derive their power from changes in the motion of the working fluid. The pressure in a turbine might be infinite, yet unless the steam possessed the requisite velocity the turbine would not act. On the other hand, if the steam possessed the requisite velocity then the pressure might be—if that were practicable—absolutely zero, and yet the turbine would work quite normally. The statical steam pressure acts equally on the back and front of each vane, and hence produces neither end thrust nor rotation. As the steam passes through the turbine the direction of its motion is altered by the vanes, and hence these vanes must exert force on the steam. It is this force, or, rather, the corresponding reaction of the steam on the vanes, which causes the rotation and where, as in the Parsons turbine, the vanes are not symmetrical, an end thrust.—From "The Theory of Steam Turbines," by Frank Foster, in *The Engineering Review*, London, May, 1904.

Flow of Steam Through Nozzles.—The illustration, Fig. 6, shows the effects of nozzles of different shapes upon steam flowing through them. The general form of the jets issuing from the nozzles is substantially as shown by Strickland L. Kneass, engi-

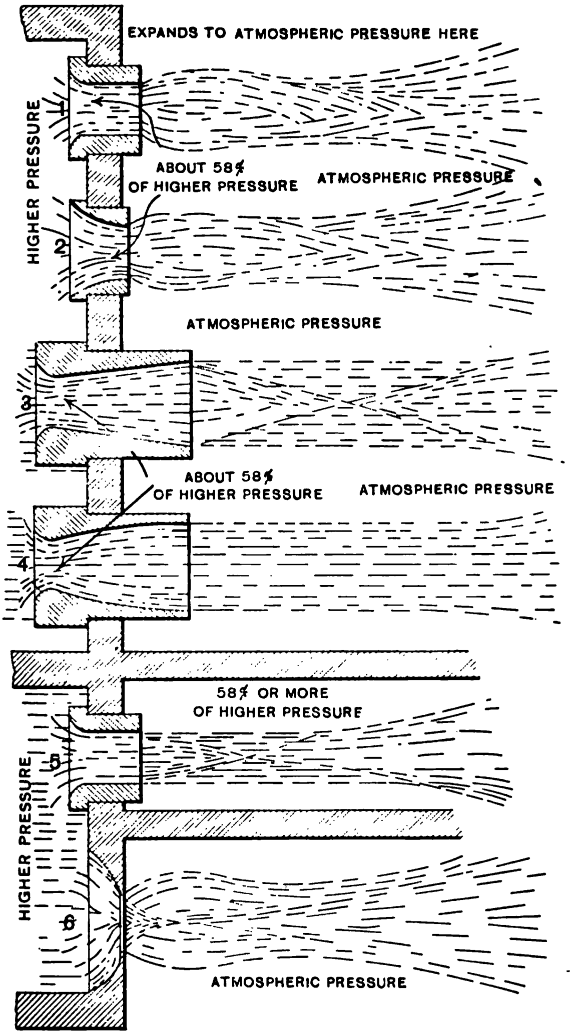


Fig. 6. Types of Steam Nozzles and the Shapes of Jets Discharging from them.

neer of the injector department, William Sellers & Co., inc., Philadelphia, in his "Practice and Theory of the Injector."

In Fig. 6, five styles of mouthpieces are illustrated, and with the exception of No. 5, steam is supposed to be flowing from some higher pressure down to atmospheric pressure. In No. 5, steam is assumed to discharge into a closed chamber in which the pressure is maintained at the "critical" pressure referred to below. The five styles of mouthpieces are as follows:

1. Short cylindrical tube, slightly rounded inlet.
2. Nozzle with converging walls—such as would be used to produce a solid water jet.
3. Diverging nozzle with rounded inlet and straight taper sides.
4. Diverging nozzle, rounded inlet and diverging sides, curved as shown.
5. Same as No. 1.
6. Orifice in a thin plate.

While the nozzles shown are supposed to be of a circular cross section, it is evident that it is the area of the section which is of importance and not the shape of the section. A rectangular-shaped section would answer as well as a circular section, for all practical purposes, and in fact is oftener used for the steam passages between the vanes of turbines.

A peculiarity of the flow of steam through nozzles is that the absolute pressure in the throat of the nozzle does not vary greatly from 58 per cent of the absolute initial pressure, unless the absolute pressure at discharge should be more than 58 per cent of the initial pressure. It can be shown that theoretically this should be the case, although it is known from tests that the pressure may be somewhat more or less than this. The pressure of 58 per cent of the initial pressure is called the *critical* pressure and has an important bearing upon the subject of the flow of steam.

Nozzles with Parallel Walls or Converging Walls.—Many experiments have been conducted on the flow of steam through nozzles like Nos. 1 and 2 and it is found that when discharging into a medium which has not more than 58 per cent of the initial pressure (equal to the critical pressure) the velocity of discharge is not far from the constant value of 1,450 feet per second and the

weight discharged is also a constant quantity. Suppose that one of these nozzles were arranged to discharge into a tank, the pressure in which could be controlled by a valve, and that at the start the tank pressure was almost equal to the initial pressure, but was gradually decreased until it finally became zero. We should find that the velocity and weight of discharge from the nozzle gradually increased, as the difference in pressure increased, until the point was reached where the tank pressure became 58 per cent of the initial pressure. At that instant the velocity of discharge would be between 1,400 and 1,500 feet per second and any further lowering of the pressure in the tank would have no effect on the velocity or weight discharged. The discharge would remain at the same rate per second, however low the final pressure became, even if carried down to vacuum.

In nozzles like Nos. 1 and 2 the pressure of the steam does not drop much, if any, below the throat pressure until the outlet is reached, but when the steam issues from the nozzle its pressure suddenly falls to that of the medium outside. If discharging into the atmosphere it would fall in pounds an amount equal to 42 per cent of the initial pressure. In making this sudden drop the steam is free to expand in all directions, the jet enlarges after issuing, becomes broken up and scattered and is inefficient for turbine propulsion. In this, as in all cases where the velocity of a jet is checked, its kinetic energy is converted back into heat, generated by the friction and eddying of the steam, which tends to superheat the steam at discharge.

Nozzles for Complete Expansion.—Where there are great differences of pressure, a more efficient steam jet can be secured by using a diverging nozzle, like No. 3, such as has been employed for many years in steam injectors. In such a nozzle the steam expands to the lower pressure within the nozzle itself and when the steam discharges it takes the form of a solid cylindrical jet, equal in diameter to the outlet diameter of the nozzle. A diverging, or more properly a converging-diverging nozzle, has a converging inlet, like nozzle No. 2, to which is added a diverging outlet, for the purpose of controlling the expansion of the steam beyond the throat of the nozzle. The pressure at the throat is about the same as the pressure in nozzles 1 and 2, or 58 per cent of

the initial pressure. Beyond the throat the cross-sectional area increases just sufficiently to accommodate the rapidly increasing specific volume of the steam (space occupied by unit weight) which occurs as the pressure drops. When thus proportioned, the walls restrict the expansion of the steam in a lateral direction, but allow free expansion longitudinally. Such a nozzle is known as an expansion nozzle and by its use the full expansive force of the steam is utilized and a very high velocity of outflow attained. A nozzle like No. 4 has been found to give slightly better results than one with straight, conical sides, like No. 3, but it is more difficult to construct.

Complete expansion may be obtained in a straight or converging nozzle by arranging so that the pressure of the medium into which it discharges shall not be less than 58 per cent of the higher pressure. Under these conditions the steam will issue from the nozzle in straight, parallel lines, or nearly so, since there is no excess internal pressure to make the jet bulge, and all the expansive force of the steam will be expended in giving velocity to the jet. In the illustration, nozzle No. 5 is intended to show that complete expansion may be realized in a straight nozzle discharging into a tank in which the pressure is 58 per cent of the higher pressure.

Orifice in Thin Plate.—When steam issues from an orifice in a thin plate, as in nozzle No. 6, the swelling of the jet after leaving the opening is even more marked than in the first two nozzles shown. This is because the internal pressure of the steam is higher in the orifice in the plate than at the mouth of the tubes in the first two examples. The steam has an opportunity to expand more fully before leaving the tubes than it does in passing through an orifice in a thin plate.

Distinction Between Impulse and Reaction Turbines.

Impulse Steam Turbine.—Both water and steam turbines are grouped into two general classes known as impulse and reaction turbines, or better, into action and reaction turbines. These terms are somewhat misleading, however, because all practical turbines operate both by the action and reaction of the working fluid and it would be clearer to designate the two types in some other way.

In Fig. 7 is a simple impulse turbine having curved vanes against which the jet of steam impinges. The expansion of the steam is completed within the nozzle, and there is no expansion in passing through the wheel passages. The pressure between the vanes is the same as the pressure within the casing in which the wheel runs and the steam flows freely through the wheel passages in virtue of the kinetic energy given it in the nozzle. The wheel is driven ahead, first by the pressure due to the impulse of the steam and then, after the vanes have reversed the direction of flow, by the reaction of the steam.

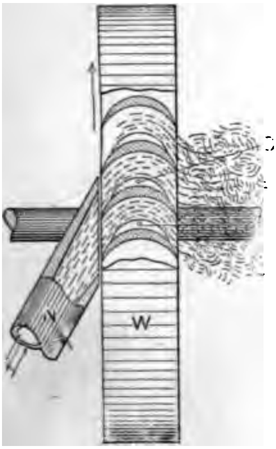


Fig. 7. Usual Type of Impulse Wheel.

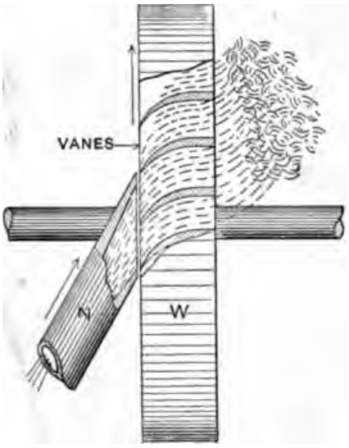


Fig. 8. Impulse Wheel in which there is no Reaction.

In Fig. 7, *N* is the nozzle, which may or may not be a diverging nozzle, according to the pressure against which it is discharging, and *W* is the wheel. With the vanes constructed as shown, there would be spaces *S S* not filled by the steam, since the area of the passages at these points is greater than at the entrance and exit. Some manufacturers, however, make the blades thicker at the center than near the edges, to maintain a constant area and so avoid possible eddy currents. Although a so-called impulse wheel, it will be evident that this wheel acts both by impulse and reaction. The chief characteristic of this type is that the expansion occurs wholly in the nozzle or guide passages, as the case may be.

What would *strictly* be an impulse wheel is shown in Fig. 8, where the vanes are so curved that with the wheel held stationary the steam would leave them in a direction parallel with the shaft. The wheel is therefore propelled solely by the pressure against the vanes due to the impulse of the steam, and would be inefficient because the steam would have a high residual velocity when it left the wheel. Such a wheel is on the principle of the stationary vane in Fig. 4, against which the water exerted only half the pressure that it did when the force of reaction was taken advantage of.

Reaction Steam Turbines.—In Fig. 9 is the simplest type of

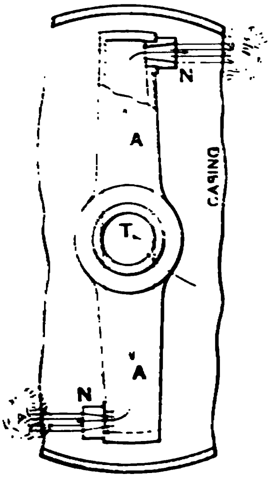


Fig. 9.

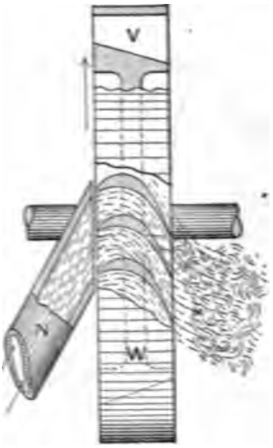


Fig. 10.

Two forms of Reaction Wheel.

wheel. Here the steam enters the trunnion T, flowing outward through the two hollow arms A A, until it dis- through the nozzles N N. The arms therefore rotate in tion opposite to that in which the steam escapes, and are entirely by the reaction of the steam. The chief difficulty is type of wheel is the excessively high speed of rotation. osing the wheel to be perfectly free to move, its momentum ld be equal to the momentum of the escaping steam. The ocity of the arms would theoretically be less than the velocity the steam only in so far as the mass of the arms was greater han that of the steam.

In Fig. 10 is shown a more practical form of reaction wheel. Here there is first the impact of the steam against the buckets, but the expansion in the nozzle is only partial and the steam expands still more and acquires additional velocity in flowing through the wheel, provision being made for this, if necessary, by having the passages diverge in the direction of the flow, as shown at V. The steam therefore reacts upon the wheel when it leaves the vanes as a result of the energy acquired in the wheel itself, and this feature gives it the name of a reaction wheel. It will be seen, however, that the wheel acts both by the impulse and the reaction of the steam just as in the case of the impulse wheel. *The distinction between the two is that in the impulse wheel the expansion of the steam is complete within the nozzle and in the reaction wheel it is not completed until after it enters the wheel passage.* If it were possible to attach a steam gauge to one of the spaces between two wheel vanes, it would show a pressure equal to the pressure of the medium in which the wheel was turning in the case of the impulse turbine and a pressure higher than that of the surrounding medium in the case of the reaction turbine.

Shape of Vanes in Impulse and Reaction Turbines.— The illustrations,* Figs. 11 and 12, represent the guide vanes and moving

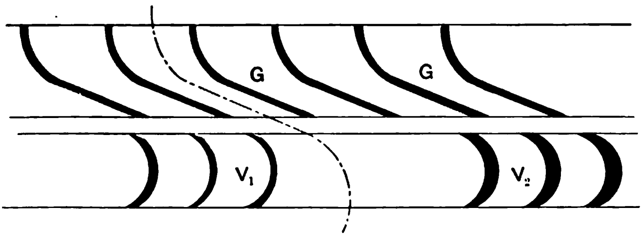


Fig. 11. Shape of Vanes in Impulse Wheels.

vanes of impulse and reaction turbines respectively. In impulse turbines the wheel vanes are symmetrical or nearly so. In Fig. 11, G G are the guide vanes which, if the fall of pressure is small, need not provide diverging passages. The wheel vanes at V_1 have both faces parallel, and the vanes at V_2 are thicker at the

*From a paper by M. J. Rey read before the Sociétié de Ingenéuirs Civils de France, March, 1904.

center than at the edges, forming passages of a uniform width, as explained in connection with Fig. 7.

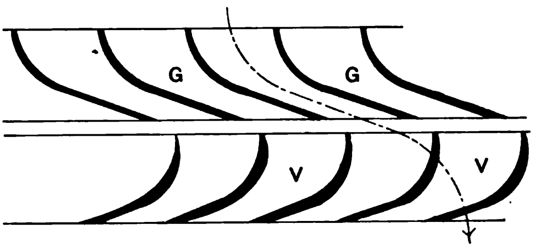


Fig. 12. Shape of Vanes in Reaction Wheels.

In Fig. 12 *G G* are the guide vanes and *V V* the wheel vanes showing in a general way the contour that must be obtained in reaction turbines.

Steam Turbine Types.

The Simple Impulse Turbine.—The simplest possible arrangement of the steam turbine is shown in the diagramatic sketch, Fig. 13, where *N* is a nozzle directing a jet of steam against the vanes of a single wheel *W* inclosed in a casing. This is like the De Laval turbine, which must utilize steam flowing with a velocity of 3,000 to 4,000 feet a second, and the peripheral velocity of the wheel should be nearly one-half of this to utilize the total energy of the steam. In the De Laval turbine the peripheral velocity is frequently as high as 1,200 feet a second or as high as safety will permit with the strongest materials for the rotating member. Such high velocity of rotation makes it necessary to use speed-reducing gearing.

Principle of the Compound Turbine.—In turbines of large size it is desirable, and in fact necessary, to avoid such high speeds of rotation and to do away with the reducing gears, and this is accomplished in several other types of turbines through compounding. A compound turbine may be built either for water or steam, and it is entirely possible for a water jet to flow at such great velocity as to make compounding desirable for a water turbine. The principle of compounding is very simple and is thus explained in Bodmer's text book "Hydraulic Motors":

"If a turbine is allowed to run at a much lower speed than at

the best, the water leaves the buckets with a very considerable absolute velocity, and there is consequent loss from unutilized energy. This energy might, however, be usefully employed in driving a second turbine, the water, after leaving the first, being deflected by a set of stationary guide vanes to cause it to enter the

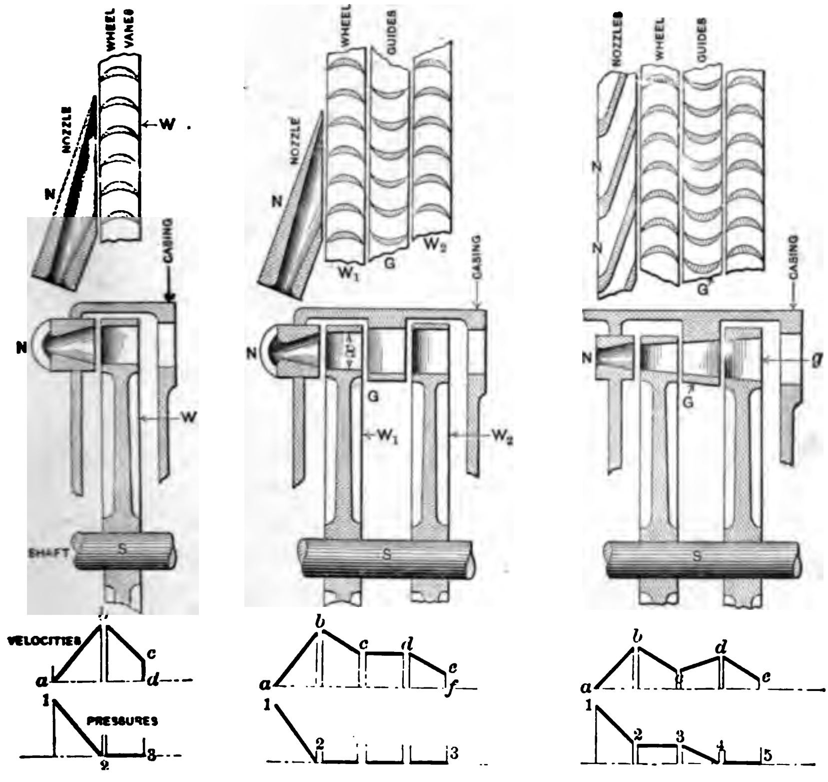


Fig. 13. Simple Impulse—De Laval Type.

Fig. 14. Compound Impulse —Riedler-Stumpf Type.

Fig. 15. Compound Impulse —Curtis Type.

Steam Turbine Types.

second wheel at the proper angle. Both turbines could be keyed to the same shaft and their speed would be much lower than that of a single turbine driven by the same head of water and utilizing it to the same extent. This arrangement would constitute a compound turbine and it is clear that, instead of two wheels only, three or more might be employed in the same way, the speed being

lower the greater the number of wheels. The only object in using a compound turbine in preference to a single one would be to reduce the speed in cases where the head was great and high velocity of rotation inconvenient or impracticable."

Compound Impulse Turbines.—There are different methods of taking advantage of this principle of compounding, the simplest one of which is shown in Fig. 14. Here steam flows through the expansion nozzle N which reduces its pressure to that of the medium in which the turbine wheels rotate. The steam then impinges against the vanes of the wheel W_1. Its direction is then reversed by the guide vanes G and it next impinges against the vanes of wheel W_2, on the same shaft S as the other wheel. In this example the action is the same as outlined above by Bodmer, and the steam, having acquired its velocity in the nozzle, flows through the passages of the turbine in virtue of its inertia. This plan has been carried out in the turbines of Professors Riedler and Stumpf, although the arrangement of the parts is different.

At the bottom of the engravings Figs. 13 to 17, are diagrams, the upper ones of which show the change in velocity and the lower ones the change in pressure of the steam in the different steps of its progress. In the first illustration the velocity of the steam increases from a to b, and as it flows through the nozzle most of this velocity is absorbed by the wheel, the height of the line c d indicating the residual or unused velocity of the steam as it leaves the wheel. The lines 1-2 and 2-3 show that the pressure drops in the nozzle but does not change after the steam strikes the wheel.

In the second illustration, Fig. 14, the velocity increases in the nozzle, but in the guide passages where no work is done it remains constant, or would do so except for frictional losses, and finally in the last wheel the velocity drops to the point e. The pressures, however, indicated by 1 2-3 are as in the previous case.

Proportions of Passages in a Compound Turbine.—It will be noticed that the height x of the passage through the first wheel W_1, in Fig. 14, is the same as the diameter of the nozzle, while the height of the passages in the guide G and the second wheel W'_2 is slightly greater. The drawing is made in this way to

emphasize one of the principles of the flow of fluids through a turbine. In flowing through wheel W_1, part of the steam's velocity is given up to the wheel so that the velocity in guide G is less and more room must be allowed for the slower moving fluids. No increase in the width of the passage in wheel W_1, is required, because, while the absolute velocity of the steam decreases as it progresses through the wheel, the velocity relative to the moving vanes does not change, neglecting frictional losses. For the same reason, also, the passages through W_2 are the same in height as the guide passages G. As a matter of fact a turbine, equipped with nozzles like Fig. 14, would not require the passages to be proportioned as above outlined, because the stream of fluid would have ample room by spreading out around the periphery as it approached the exhaust end.

Modified Form of Compound Impulse Turbine.—Difficulty is experienced in making steam flow through groups of irregular passages without any impelling force to overcome friction other than its own inertia. To supply the necessary impelling force to make up for frictional losses the modification shown in Fig. 15 has been used, notably in the Curtis turbine. The steam as before flows through diverging passages N N and impinges against the vanes of the first wheel and then passes through the guide vanes G to the second wheel. The depth of these passages increases slightly from the point where the steam strikes the first wheel to the point where it leaves the last wheel at g. A part of the expansion, however, is reserved to take place in the wheel and guide vanes for the purpose of accelerating the velocity sufficiently to compensate for its retardation by friction, although most of its expansion occurs in the nozzles as before. This turbine, like the two previous ones, is essentially an impulse turbine, though the pressure in the wheel passages is slightly higher than that of the medium in which the wheels rotate.

Stage Turbines.—The arrangement of Fig. 15 is sometimes modified by having two or more groups of wheels and guides, each of which is in a separate compartment so that the reduction in velocity and pressure is not as great in any one step. This plan is followed in the Curtis turbine, where each compartment

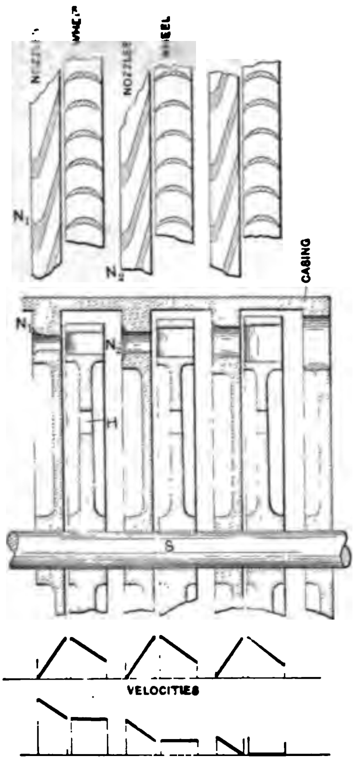


Fig. 16. Compound Impulse—Multi-cellular or Rateau Type.

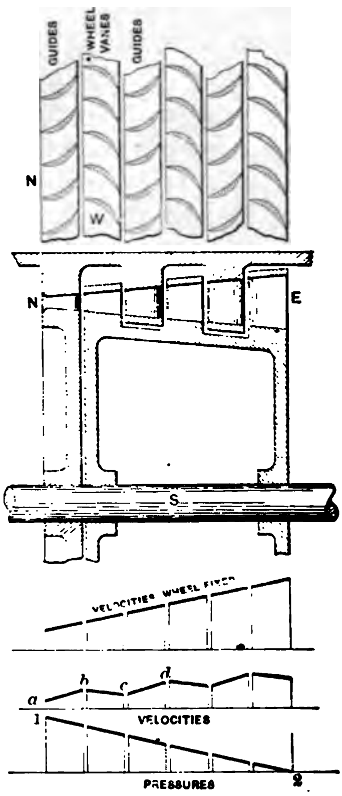


Fig. 17. Compound Reaction—Parsons Type.

Steam Turbine Types.

in which the pressure is stepped down is called a stage, and it is known as a multi-stage turbine.

The simplest form of stage turbine is shown in Fig. 16, which illustrates the principle of turbines like the Rateau and Zoelly. This is in effect a series of simple turbines like that of Fig. 13, each of which is in a separate compartment. There are usually enough compartments so that the steam does not have to drop more than .4 of its initial pressure when flowing from one compartment to the next, and diverging nozzles are not required. Steam enters through the guide passages N_1 and expands down

to the pressure in the first compartment. The pressure within the wheel passages is the same as without and to insure a uniform pressure in all parts of the compartment holes H are sometimes made through the wheel disks. The second set of guide passages N_2 has a larger area than the first to accommodate the increased volume of the steam and the same is true of the succeeding passages. The velocity diagram below shows that there is a small unused residual velocity in each case and the pressure diagram shows how the pressures are gradually stepped down.

Compound Reaction Turbines.—We now come to the reaction turbine of which the Parsons turbine is the most prominent representative. In this type the steam expands continuously from boiler pressure to vacuum. There are alternate rows of guides and vanes, the latter being attached to the drum on the turbine shaft. The steam flows through a fixed ring of directing blades N onto a revolving ring of similar blades W and so on, its pressure being reduced a few pounds, say two or three, at each step. The steam finally discharges at E. It will be seen that if the wheel were fixed and the steam allowed to flow through the turbine the passages themselves taken together would constitute a large expansion nozzle, and the flow of the steam would increase from beginning to end as shown in the upper diagram placed beneath the section of the wheel.

Let the wheel rotate, however, and the velocity acquired in passing through the first guide ring would be partially absorbed by the first wheel; and the velocity acquired in the next ring of guide blades would be partially absorbed in the second wheel, and so on. The line of velocities, therefore, would be represented by *a b c d*, etc., in the middle diagram. The pressure, however, drops gradually from beginning to end as represented in the last diagram.

This diagram shows that the pressure in the passages of the turbine is maintained higher than the final pressure, which, as has been explained, is the characteristic of the reaction principle.

In subsequent chapters modifications of these simple types will be shown, but what has been given is believed to be sufficient to enable the reader to understand the descriptions of the various turbines which follow.

CHAPTER II

EARLY STEAM TURBINE PATENTS.

It is probable that the first steam engine was a turbine. In Hero's "Spiritalia," a book on pneumatics issued in the second or third century, is a description of the whirling eolipile consisting of a small hollow sphere mounted on trunnions, one of which is hollow for the admission of steam. The sphere is caused to rotate by the reaction of steam flowing from two diametrically opposite nozzles having bent mouthpieces. This is frequently spoken of as the beginning of the reaction turbine; and to Branca, who issued a work, entitled "The Machine," published at Rome in 1629, is given credit for the first impulse wheel. This volume contains an illustration of an eolipile, in the form of a negro's head, placed over a fire. A blast of steam proceeds from the mouth and impinges against the blades of a large wheel which it was proposed to connect by means of cog wheels with a crude stamping mill for pulverizing drugs. These very early efforts could have been nothing more than visionary schemes, but they are scarcely less impracticable than many of the later inventions to be found in the pages of the patent records. Comparatively few of the steam turbine inventions embody even the first elements of success, probably because most of those who have directed their attention to the subject have failed to understand either what was required or what means must be taken to accomplish good results.

In selecting from among the great number of turbine patents those that appear to have useful features, the author has had in mind the requirements of the successful steam turbine as outlined in the first chapter, and has not given space to inventions unless they seemed to embody at least one feature that would contribute toward a practical and operative machine. With the exception of

*In writing this review the author has drawn on the historical material in the valuable series "Roues et Turbines a Vapeur," by M. Sosnowki, published in the August, September, October and November, 1896, numbers of the "Bulletin de la Société d'Encouragement pour l'Industrie Nationale," Paris. He has also been materially assisted in his search of the patent records by the list of English turbine patents in Neilson's treatise, "The Steam Turbine"; and by a similar list of United States patents kindly supplied by Mr. Robert A. McKee, mechanical engineer, steam turbine department, Allis-Chalmers Company.

a very few patents taken from the French patent records, specifications of the inventions mentioned are to be found either in the English or in the United States patent records.

Real and Pichon, 1827.—This machine operates by impulse and is one of the earliest attempts to produce a wheel to run at moderate speed and at the same time utilize a large percentage of the energy of the steam by the principle of compounding. Certain of

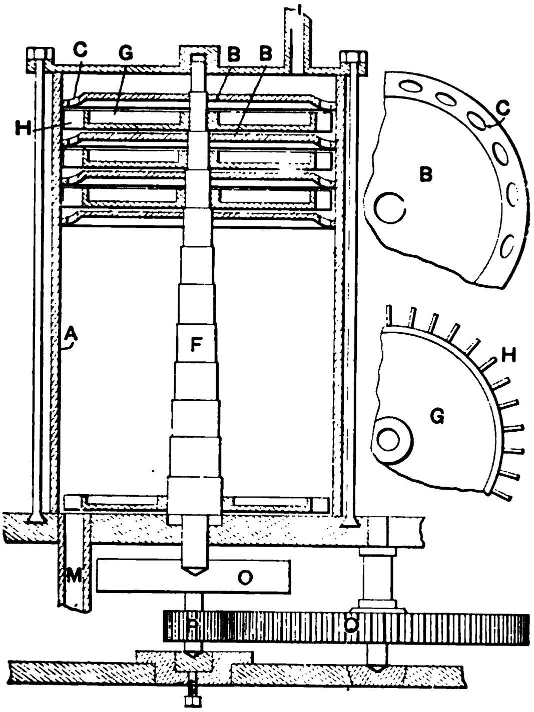


Fig. 1. Real and Pichon Compound Turbine.

the details of the original patent drawing are somewhat obscure, but the illustration has been made to correspond with the text as nearly as possible. The cylinder *A* contains a succession of disks, *B*, which divide the cylinder into compartments. The shaft *F* is turned with a series of steps upon each of which is carried a turbine wheel *G*, having short radial blades, *H*, around its periphery. Steam is admitted from the boiler through the pipe *I* at

the top into the first compartment and flows in the form of jets through a series of openings, *C*, against the blades of the first wheel which runs in the second compartment. The steam next passes through a second series of holes in the second disk and impinges against the second wheel and so on to the bottom of the cylinder, where the steam exhausts through the pipe *M*. The shaft and wheels are carried by a step bearing and power is supposed to be transmitted through the gears *P* and *Q*. The openings, *C*, in the circumference of the disks, *B*, are bored obliquely, so the steam will impinge as directly as possible against the faces of the blades. With this plan the pressure will drop only a few pounds from chamber to chamber, giving the steam a comparatively low velocity of flow.

Avery Turbine, 1831.—The first steam turbine patent to be issued in the United States was to Foster & Avery for a reaction wheel of the Hero type. Strangely enough, this is one of the few turbine inventions that has been developed and put into actual use, and probably it is the only steam turbine used in commercial work

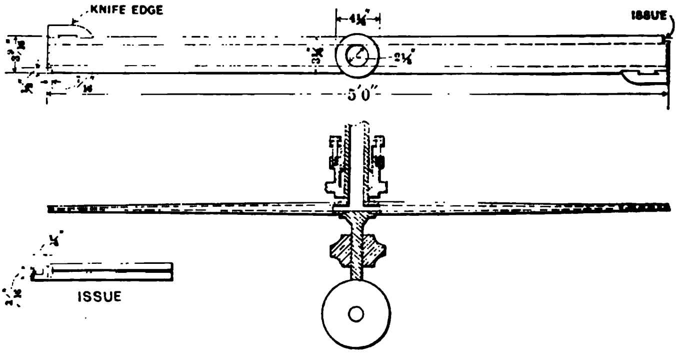


Fig. 2. Avery Reaction Wheel.

in this country until a considerably later date. There were several of these machines in operation in 1835, some of which were used to drive saw mills near Syracuse, N. Y. In 1901, Prof. John E. Sweet contributed a description to the Transactions of the American Society of Mechanical Engineers, accompanying it by a sketch made from an original drawing of the Avery wheel, reproduced in Fig. 2.

The arm is made, with the exception of the end pieces and knife blades, of two pieces of iron brazed together from end to end at the edges. The openings at the ends of the arms for the steam jets were ⅛ by ¼ inch. The speed of the tips of the arms was, of course, enormous. Mr. Avery states in his notebook that the speed of the arms of a 7-foot wheel placed upon a locomotive in 1836, which was put upon a railroad near Newark, N. J., and ended its life in a ditch, was at one time 14½ miles a minute at the periphery. A difficulty met with was the end pressure on the hollow shaft, which was overcome by running the end of the shaft against the edge of a wheel set at right angles. The trouble in setting up the packing around the hollow shaft became a serious matter. It was also found that the knife edges at the end of each arm were cut away by the steam and required frequent renewal. The noise also was very objectionable.*

Leroy, 1838.—In commenting on Avery's invention Prof. John E. Sweet has said that he long had the conviction that expanding nozzles applied to the Avery turbine, in place of the plain orifices used, would give the benefit of expansion and produce superior results.

*In a paper presented by John Richards before the Technical Society of the Pacific coast in 1904 and published in the Journal of the Association of Engineering Societies for September, 1904, is the following account of the Avery engines, written by Professor Sweet, a near relative of Mr. Avery: "In respect to the history of the Avery engines, these were made 75 to 80 years ago by William Avery, a local mechanic in Syracuse. There were about 50 constructed and put in use. One of the runners is now in my possession; another, that I saw years ago, had a hollow shaft of perhaps 1¼-inch bore. The head or runner was of sword shape, the arm 1 by 3 inches at the center and ⅜ by 3½ inches at the ends, the diameter swept being about 5 feet. Steam was admitted through the shaft by means of a stuffing box, passed through the shaft to the hollow arms and escaped at a tangential issue ⅛ inch by ¼ inch, at the rear corners of each arm, the ends of which were stopped by plugs brazed in. Owing to the rapid rotation of the arms—10 to 15 miles per minute—the front edges were so rapidly cut away that replaceable blades made of tempered steel were inserted so they could be renewed. The fact that the engine had to be taken to a blacksmith shop every 3 or 4 months for renewal or repairs had more to do with its abandonment than its lack of economy. As to the latter, people who knew the facts, or claimed to do so, said that when they changed to the common slide-valve engines there was no gain in steam economy over the Avery engine. Another feature that worked against the Avery engine was the stuffing box around the shaft, which in the hands of workmen of that time was apt to be set up so as to consume a large part of the power in friction. This was a natural consequence, as the wear was rapid. What the result would have been with a truly ground shaft in a metal bush, instead of a turned shaft and stuffing box, making the issues expanding nozzles and multiple expanding by 2 or 3 arms in separate cases and connecting to a condenser, is not known. It might rival a pretty good modern engine, if not the best. The Avery engines were used in saw mills and wood-working shops of the time. They had weak starting power, and did not need much for the uses named. They ran at such a fearful speed that the reducing motion was an impediment. Mr. Avery had to employ bands, which were far more objectionable than gear wheels."

Leroy is perhaps the first on record with this idea of the application of the expanding nozzle. He was a prolific inventor and had definite notions about many features now employed in turbines. Figs. 3, 4 and 5 show three styles of rotating arms that he proposed for reaction wheels. The nozzle at N is clearly a diverging nozzle, as are also the orifices in Fig. 5. It is uncertain, however, whether he understood the principle of the diverging nozzle, because he states in one place that a nozzle in the form of a tube, Fig. 4, will produce a higher steam velocity than a funnel-shaped opening. This would be true if the funnel flared too much, as

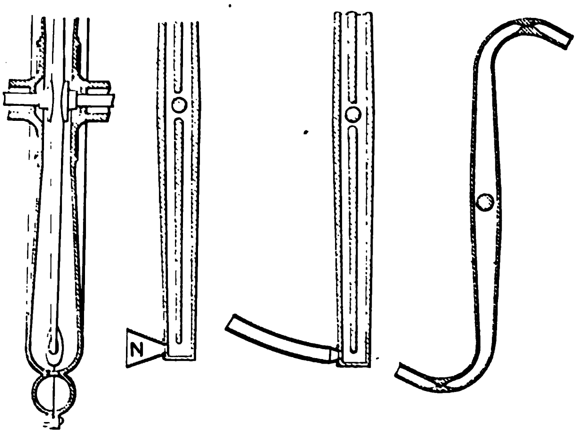


Fig. 3. Fig. 4. Fig. 5.

LeRoy's Reaction Wheels.

seems to be the case. It is a curious fact that the author has learned of recent experiments by a mechanical engineer who is at work upon the turbine problem, which show the same result. When a nozzle flares too much the expansion of the steam is completed before the end of the nozzle is reached, and the effect of the divergence beyond that point is to check the flow of velocity just as is the case in a water nozzle which diverges.

Leroy was one of the first to propose a compound turbine. He shows two illustrations of machines—one a reaction and one an impulse turbine, in which each wheel is encased in a separate chamber. In the reaction turbine steam enters the hollow arms of the first wheel through a trunnion at the center and escapes

through openings in the periphery into the first chamber. It is then conducted by a pipe to the second wheel in a similar manner, where it finally escapes into the second chamber, and so on. His compound impulse turbine is entirely similar in principle to the Real and Pichon turbine except that instead of a succession of openings for the steam around the periphery the steam is conducted to each wheel by a single pipe. His drawings of the compound turbine are unpractical, because he makes no provision for the increasing volume of the steam as it expands. The drawings show passages of the same area near the exhaust end of the turbine as at the inlet end.

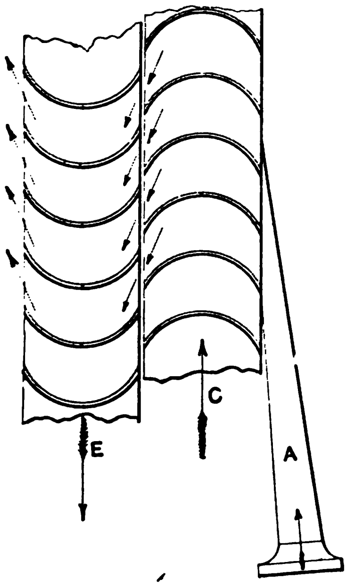


Fig. 6.

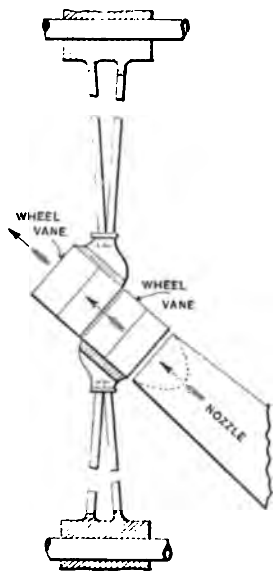


Fig. 7.

Pilbrow's: Wheels rotate in opposite directions.

Another early inventor who attempted to use the diverging nozzle was Von Rathen, who, in 1847, invented a reaction wheel having conical-shaped mouthpieces through which the steam escaped. These also flared so much as to be a hindrance rather than a help.

Pilbrow, 1842.—The inventions of Pilbrow were numerous. He

experimented on the flow of steam and determined that for economical results the peripheral velocity of the wheel must be very high, and accordingly devised various arrangements for compounding with a view to reducing the velocity to a practical rate. In all his compound turbines, however, he adopted the plan of running two or more wheels in opposite directions without stationary guide vanes, as shown in Fig. 6. Here steam enters through the nozzle, *A*, impinging against the blades of wheel *C*, which rotates in the direction of the arrow. The steam then passes through this wheel and discharges against the blades of a second wheel, *E*, rotating in the opposite direction. Fig. 8 shows how he proposed to carry the idea still further by using several wheels, the alternate wheels rotating in opposite directions. Still another construction

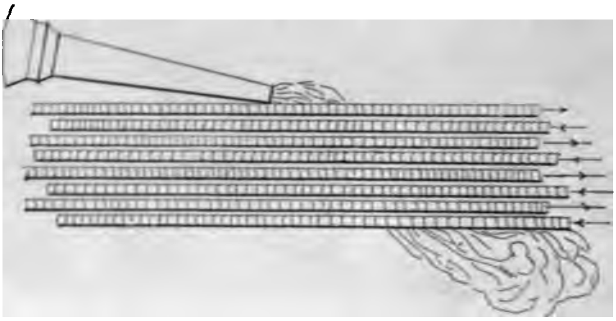


Fig. 8. Pilbrow's Multi-Wheel Turbine.

that he proposed is indicated in Fig. 7, where the two wheels rotate on parallel shafts, as shown, and have inclined vanes so located that steam from the nozzle will flow through the vanes of both wheels in the direction of the arrows. The buckets are curved, as in Fig. 6, and the wheels, of course, rotate in opposite directions.

Another interesting invention of Pilbrow is illustrated in Fig. 9. This is a reversing turbine arranged with a number of nozzles that can be shut off or opened successively by means of a rotary valve. The plan of using several nozzles, which are brought into or out of action by valves, as used in the De Laval and Curtis turbines, probably here has its introduction, and the invention is of value on this account. Steam enters the chamber, *C*, in which is located a

rotating segment that covers or uncovers the nozzle openings, *a, b, c,* etc. At *A* is the wheel with vanes pointing in one direction and at *B* one with vanes in the opposite direction. Half the nozzles connecting with chamber *C* direct the flow of steam against wheel *A* and the other half against wheel *B*. By rotating the segment, steam can be admitted to either wheel, causing the turbine to revolve in either direction, as desired; and also the amount of steam admitted can be adapted to the power required. A rotating valve of this description is not to be advocated as a durable construction.

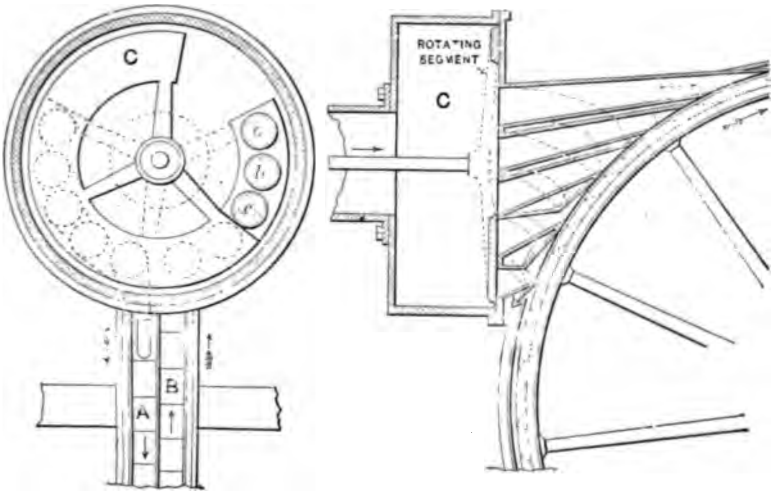


Fig. 9. Pilbrow's Plan for Reversing with Valve for Controlling Nozzles.

Wilson, 1848.—The inventions of Wilson rank among the two or three most important early steam turbine patents. His designs are the forerunners of the present Parsons type. He devised several compound reaction turbines in which the steam flowed through alternating sets of stationary and rotating rings of blades, expanding gradually during its passage through the apparatus. Fig. 10 is a sketch of his most valuable invention. Steam enters at the left, passes through the turbine in a longitudinal direction and exhausts at the outlet at the right. The vanes, *a, b* and *c,* are attached to the drum, *D,* and rotate with it, while *d, e* and *f* are stationary guide vanes. The depth of the vanes increases from inlet to outlet,

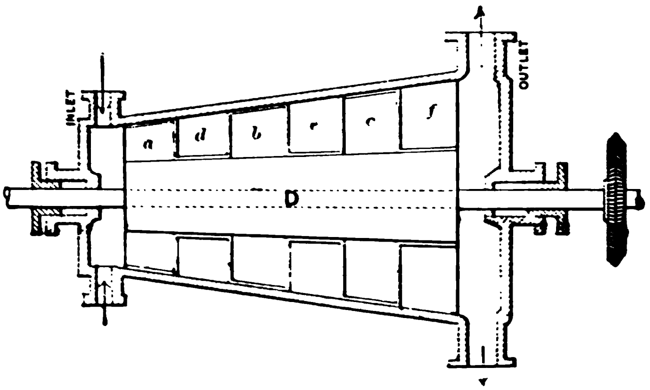


Fig. 10. Wilson's Compound Turbine.

allowing for gradual expansion of the steam. This is really the Parsons turbine reduced to its simplest elements.

Another type—the radial flow wheel—is shown in Fig. 11. Here there are alternating stationary and moving vanes, and the steam flows outwardly through them, at the same time expanding to a lower pressure.

In Fig. 12 is still another type in which there is a single rotating

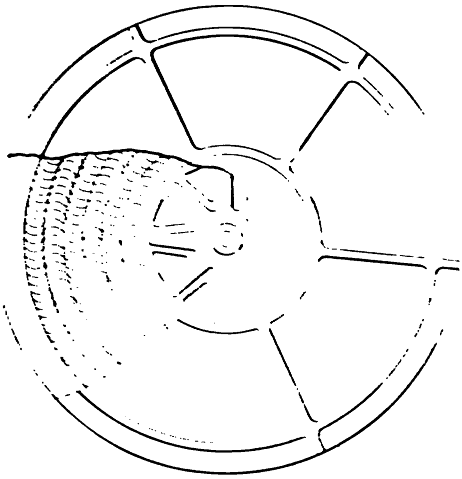


Fig. 11. Wilson's Compound Radial-flow Turbine.

ring of blades marked B. The steam is expanded and utilized upon this one ring of blades several times in succession by following a tortuous course back and forth through this ring B. Steam enters at A, passes through the moving blades to the chamber C, then returns through the guide vanes in this chamber to the chamber D; again it passes through the guide vanes to the wheel and into chamber E; then to chamber F, and so on. These successive chambers increase in size to allow for the increase in the volume of

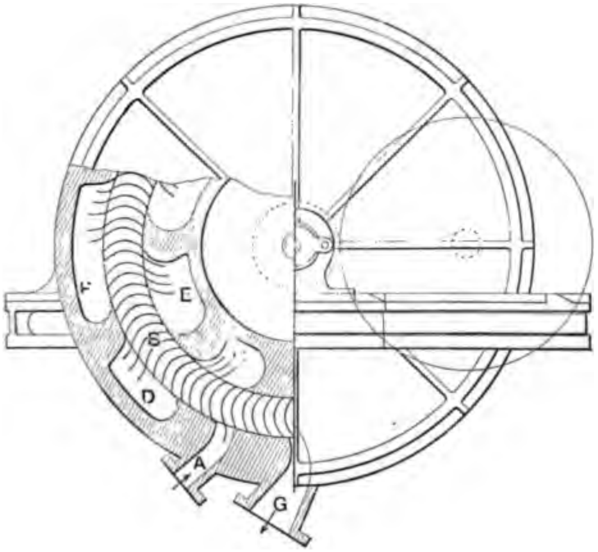


Fig. 12. Another Type of Wilson Turbine.

the steam as it progresses through the wheel, until finally it has passed around the whole circumference and exhausts at the outlet G. This plan of allowing steam to act at different points in succession on a single rotating ring of blades, has since been worked out in various other ways, as subsequent patent specifications show.

Delonchant, 1853.—The speed reduction problem was attacked by Delonchant in the same way that it was later by De Laval; that is, instead of compounding he proposed to allow his turbine to run at high speed and then used reduction gearing, in the form of the familiar "grindstone bearing." The arbor, B, of the wheel, Fig. 13,

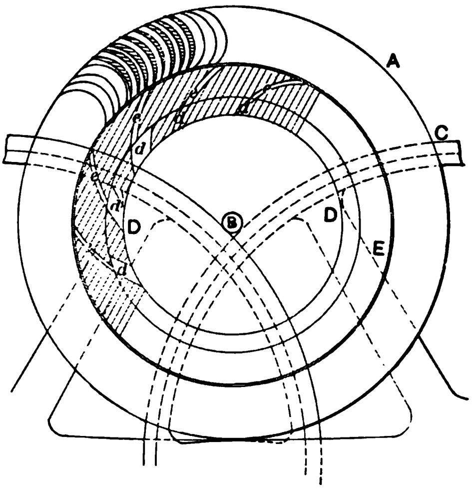


Fig. 13. Delonchant.

was supported on the circumference of antifriction wheels, *C*. In explanation he says: "By the employment of these wheels instead of ordinary bearings, not only the rubbing of the first axes will be replaced by rolling friction but power will also be transmitted to the following parts, without gearing." In the illustration, *A* is the rotating wheel; *B*, the arbor and at the center is a steam chest, *D D*, indicated in outline only. Steam passes from the steam chest through the passages *d d d;* and *E* is a ring having passages *e e e,* used in regulating the amount of steam flowing through the wheel. The passages, *d d d,* are so disposed that by rotating ring *E* the passage *e e e* through the ring will be successively cut off from the steam supply, or else opened to the supply. By moving 1/12 of a turn one passage is closed; another 1/12 closes a second passage, and so on.

Tournaire, 1853.—In this year Tournaire presented to the Academie des Sciences a paper discussing the merits of compound turbines both of the impulse and reaction types. There is a copious extract from this paper in the Bulletin de la Société d'Encouragement pour l'Industrie Nationale for September, 1896, and the facts explained by him as essential to a successful turbine are so in accordance with modern practice as to place him among the leading

inventors. He says: "To overcome the difficulties of high velocities the vapor or gas should be made to lose its pressure in a continuous and gradual manner, or by successive fractions, by causing it to react several times upon the floats of turbines conveniently situated. Since the differences of pressure are considerable it is not difficult to recognize the necessity for a large number of successive turbines in order to sufficiently annul the velocity of the fluid jet. In spite of the multiplicity of parts the device must be simple in its action and susceptible of great exactness in construction." Tournaire believed he fulfilled these conditions by means of a machine composed of several wheels, with shafts having the same axis and driving the wheel which was to transmit the motion,

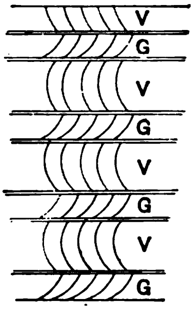


Fig. 14. Plan of Vanes in Tournaire's Turbine.

by means of pinions. A plan of the buckets and vanes is given in Fig. 14, where *G G G* are the rotating elements and *V V V* the stationary elements. He describes the construction of the turbine in detail, but these structural features are of little interest at the present time. It is to be noted, however, that he appreciated fully the necessity for expansion. He says: "As the vapor will expand in proportion as it passes from the wheel buckets and directing rings, it is necessary that the passages between them become larger and larger." He also suggests losses from leakage, saying: "A part of the fluid escaping between the spaces, which it is necessary to leave between the fixed and movable parts, will exert no action upon the turbine, nor will it be guided by the directing buckets. Shocks and eddies will be produced at the entrance and exits of

the buckets." Again, "The friction which the narrowness of the channel will render considerable will absorb an appreciable part of the theoretical work." As to the structural features he suggests among other things, that "the cogs of the pinions which will turn with great rapidity, should work very evenly without shocks and jolts," and proposes the use of helicoidal gears. His turbine, as well as some of the others already described, is a vertical turbine rotating on a vertical axis.

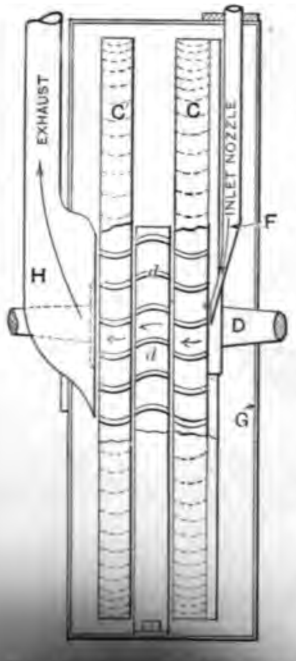


Fig. 1[illegible] [illegible]'s Compound [illegible] Turbine.

John and Ezra Hartman, 1858.—Standing in importance with the inventions of Wilson and Tournaire are the English and American patents of the Hartman brothers, from the drawings of which Fig. 15 is made. The patent relates to a "mode of obtaining motive power by causing steam or air to impinge upon a series of chambers with curved bottoms ranged around a wheel at or near the periphery thereof; and second, the general construction and arrangement of machinery or apparatus for obtaining motive power." Fig. 15 shows the most important modification of the patent, of which the following is the inventor's description: "This represents a detail of the third modification wherein we propose to employ two wheels, C C^1, both wheels being fast on one shaft, D. [illegible] the contiguous faces of these wheels for the [illegible] more returning chambers, d d, the bottoms of [illegible] in a direction opposite to that of the bottoms of [illegible] wheels. These chambers in other respects are [illegible] to those in the wheels and are fitted to a rim which [illegible] wise secured to the interior of the casing G. The [illegible] one side of the wheel and the discharge pipe, H, on [illegible] of the second wheel.

"The jet pipe on being first introduced impinges against the curved bottoms of the chambers in the wheel *C*, and is thence diverted against the fixed chambers, *d d*, whence it is again diverted on to the curved bottoms of the chambers in the second wheel, *C*, and finally passes off by the escape pipe, *H*."

Charles Monson, 1862.—We have already illustrated types of

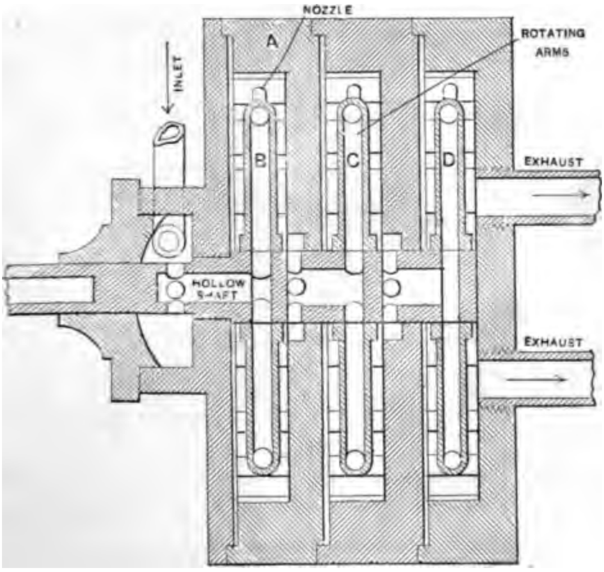


Fig. 16. Monson's Compound Reaction Wheel.

simple reaction wheels, but a search of the patent records shows that several inventors have attempted to improve on this arrangement and produce a turbine which will run at slower speed, by having a succession of simple reaction wheels, each one in a separate chamber and arranged so that steam issuing from a wheel into its chamber will then pass through to the next wheel, and so on. This in substance is the design of Monson's turbine, shown in Fig. 16. The leading specification of his patent is as follows: "A repeating rotary engine constructed in a manner so as to operate substantially as described; namely, of two or more sets of curved arms, *B, C, D*, or their mechanical equivalents; a series of two or more tight chambers or passages, *A*, and a shaft or its equivalent divided

into separate chambers and provided with induction and escape passages." The course followed by the steam will be evident from the engraving. Other patents similar to Monson's have been taken out in later years, notably by T. Barra in 1867 and by Parsons, the inventor of the Parsons steam turbine, in 1893. These latter are exactly similar in principle to Monson's and are merely construction patents.

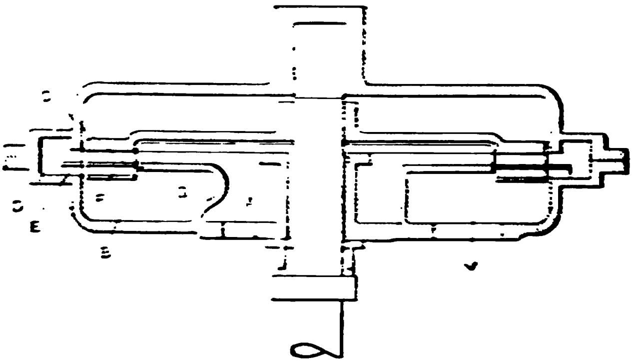


Fig. 17. Hoehl, Brakell & Gunther.

Hoehl, Brakell & Gunther, 1893.—The turbine produced by this aggregation of inventors has as its only novelty an arrangement of passages by which the steam returns on itself and so is utilized twice by the same wheel, although an additional set of wheel blades is required. Steam enters the chamber *A*, passes radially tward through the guide passages *B* to the wheel blades *C*, then charges into the annular chamber D, where its motion is reed, and escapes through *E* to another set of buckets *F*, and ly to the exhaust chamber *G*.

errigault & Farcot.—The type of turbine here exemplified is in iciple like Wilson's turbine, in Fig. 12, the latest representative which is found in the compound turbine of Messrs. Riedler and tumpf, to which reference will be made later. The inventions of errigault & Farcot took several forms, but the general principle is vell illustrated by Fig. 18 herewith. Here steam enters through he pipe or nozzle *A* and impinges against the wheel buckets, assing through to the other side of the wheel where it discharges

into pipe *B*. This pipe brings the steam around again to the inlet side of the wheel, allowing it to discharge a second time against the buckets of the same wheel, when it is again picked up by a second pipe *C*, and so on. The exhaust is finally through pipe *D*.

The arrangement consists essentially in a bundle of bent pipes having openings *a, b, c,* through which the steam impinges against the wheel buckets; and openings on the other side, *x, y,* etc., which gather up the steam flowing from the wheel and bring it round again to the inlet side. The object is to utilize the steam

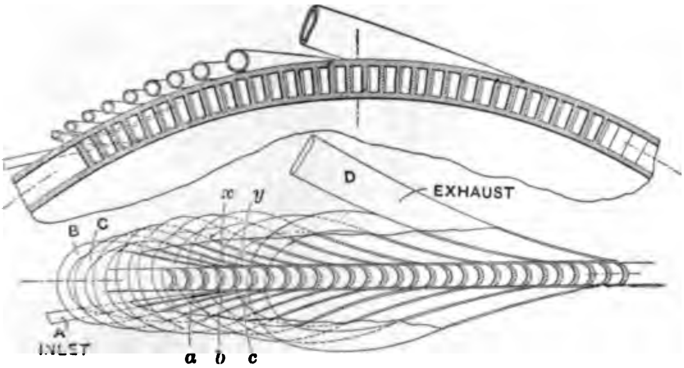


Fig. 18. Compound Turbine with Only One Wheel.

over and over without necessitating a series of rotating wheels. It is a system that has been tried with various modifications, but without much success. Its obvious disadvantages are: Large losses from friction and leakage, and the wide range of temperatures through which the same buckets must pass, thus causing condensation and reëvaporation as in the steam engine.

Moorhouse, 1877.—Among the most successful types of turbine is that having a succession of chambers in each of which is a single impulse wheel. There is only a slight drop in pressure from chamber to chamber, so that the velocity of the steam does not become excessively high at any point. The latest turbine of this description is the Hamilton-Holzwarth, built by the Hoovens, Owens, Rentschler Co., Hamilton, O., and the earliest one of which there is any record is the invention of Real & Pichon, 1827—the first patent referred to in this series. The invention of Moorhouse is for

a turbine on the same plan. As shown in Fig. 19, a, a, a, etc., are the nozzles and b, b, b, etc., the wheel buckets. Steam flows radially outward at each wheel until near the exhaust end, where there is a different arrangement, owing to a smaller drop in pressure between the successive chambers.

Morehouse realized what previous inventors of this type of compound impulse turbine had not, or at least had failed to specify, namely, that provision must be made for progressive expansion of the steam by a gradual increase in the area of the steam passages.

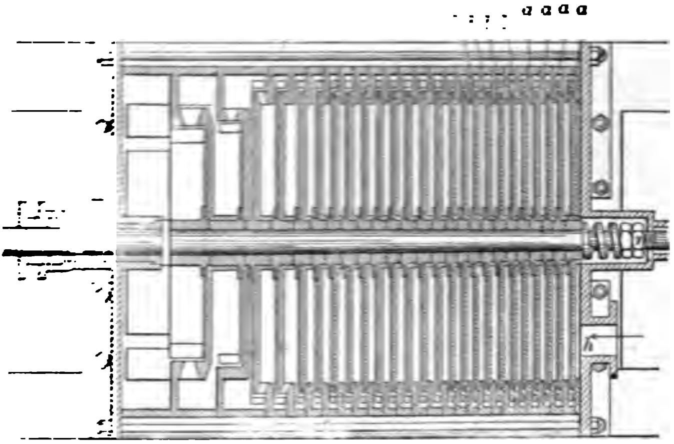


Fig. 19. Compound Impulse Turbine with Provision for Expansion of the Steam.

His patent is a broad one and fully covers this requirement as a general principle, apart from the exact method used in its application. In his specifications he says that his invention consists of a cylinder built in sections, each section composing a separate compartment. Through the center of the cylinder passes a revolving driving shaft upon which is fitted a series of turbine wheels, each having, at or near its circumference, a sufficient number of buckets. These wheels are as many in number as the compartments into which the cylinder is divided. Each compartment contains a turbine wheel, and is separated from the adjoining compartment by means of a dividing plate or diaphragm. The foregoing is condensed from his specifications, but correctly represents their mean-

He then goes on to say:

"Openings are made in the dividing plates which separate each compartment from the adjoining ones, and the area of these openings is proportioned to the pressure of the steam or other driving fluid, and to the number of compartments and turbine wheels, and to the extent to which it is desired that the driving fluid should be expanded before being finally discharged from the engine. By this means the driving fluid, admitted at its highest pressure into the smallest compartment, passes into the second compartment through

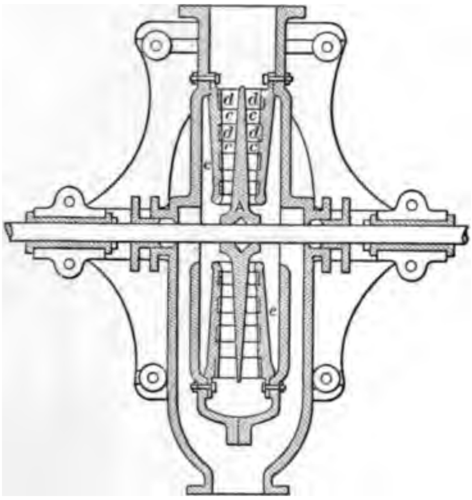


Fig. 20. Radial Outward-flow Turbine.

openings of such area that it expands to a calculated extent. The same process is repeated, etc."

He says that various forms of turbine may be used, and his first claim is as follows:

"In combination with a rotary engine, the dividing plates between the compartments provided with openings forming communications respectively of varying area between said compartments, the turbine wheels in such compartments, and a driving shaft, substantially as and for the purpose set forth."

Cutler, 1879.—This is a radial outward flow turbine in which the compound principle is used. Steam enters at the bottom, passes to the center and then flows radially outward through the passages

between the guides c c c and the wheel vanes d d d. The rotating wheel A has vanes attached to each of its two faces so the pressure is balanced on each side. Expansion of the steam is allowed for in part by the increasing width of the passages and in part by the fact that the steam is constantly flowing from a smaller to a larger diameter of wheel so that the circumferential area of the passages constantly increases from this cause as well.

[illegible].—This is another attempt to apply the compound

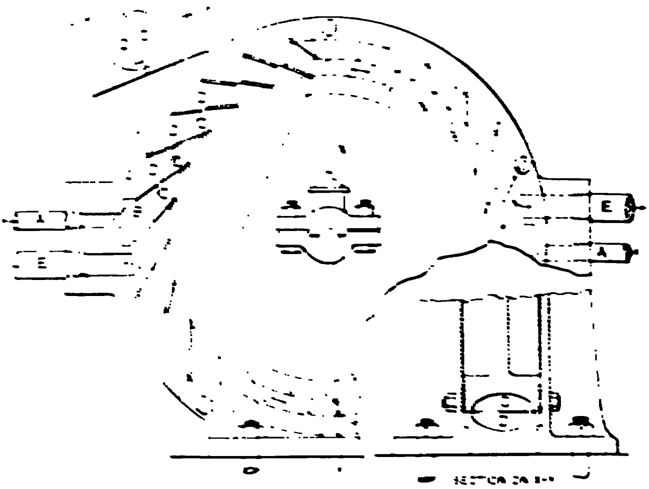


Fig. 21. Another Compound Turbine with Only One Wheel

principle to a single wheel having a single set of vanes, but differs somewhat from the turbines of Wilson and Perrigault and Farcot. Steam enters at A, passes through the nozzle and impinges against the buckets C C C. These buckets are semi-circular in shape, as indicated in the sectional view in the upper left-hand corner of the illustration. The steam enters at one side of the bucket, follows the curved surface of the bucket, and discharges into the opposite side of a semi-circular stationary bucket or guide D. Here the direction of flow of the steam is again reversed. The steam, as before, flows around the stationary guide surface and discharges [illegible]e of the buckets C, whence it is carried along to the [illegible]ionary bucket, and so on, alternately entering the succes-

sive wheel buckets C C C and the successive stationary buckets D D D. It finally discharges on the opposite side of the turbine casing, at E. In the meantime steam enters at A on the right-hand side of the casing and zigzags through the lower half of the wheel in a similar manner, exhausting at E on the left-hand side.

De Laval, 1883.—The first patent of this noted inventor was for a reaction turbine and was taken out in 1883, in several countries. According to the specifications, steam (or other fluid) enters the

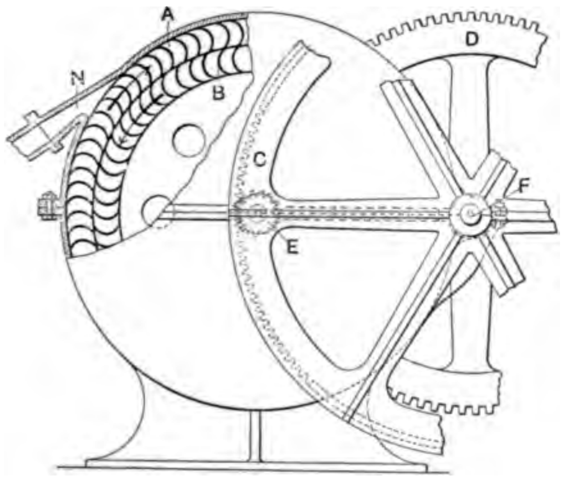


Fig. 22. Two Wheels Rotating in Opposite Directions.

wheel at the center through a nozzle, and passes outward through hollow curved arms, escaping at their ends, and causing the wheel to rotate at high velocity. The wheel shaft drives another shaft at a slower speed, by means of friction wheels, the requisite pressure between the friction surfaces being obtained by the axial thrust of the turbine wheel. The principle of this turbine is no different from that of the first American patent by Avery in 1831, but its application to centrifugal cream separators, for the extensive development of which Dr. De Laval has been responsible, was successful and marks the beginning of an important career by this inventor in the manufacture of steam turbines.

Babbitt, 1884.—B. T. Babbitt, besides acquiring fame as a

manufacturer of laundry soap, was both an inventor and a mechanic, and one of his inventions related to a steam turbine of the type shown in Fig. 22. This is an inward-flow turbine with two wheels, *A* and *B*, having rows of buckets on their peripheries. The wheels rotate in opposite directions. The steam from the nozzle *N* impinges against the buckets of the outer wheel, which it is supposed to leave with a considerable residual velocity, and gives up its remaining energy to the buckets of the inner wheel. The chief novelty of the invention is the method of transmitting power from these two wheels to the slow speed shaft *F*. The turbine wheels are mounted concentrically on two separate shafts, having the same axis, shown at *E*. There is a pinion on the outer end of each of the shafts. One of these pinions gears with the internal gear *C* on one end of the slow speed shaft *F*, and the other pinion gears with the spur wheel *D* on the other end of shaft *F*.

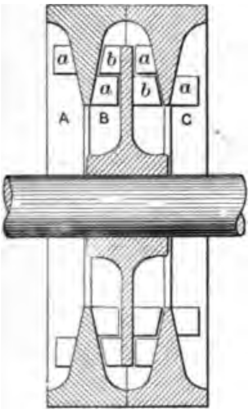


Fig. 23. Isaac Last.

Isaac Last, 1885.—This is another example where the steam is caused to return upon itself, first flowing radially outward then reversing and flowing radially inward, just as in the Hoehl, Brakell & Gunther turbine, Fig. 17. In Last's turbine, however, compounding is carried further than in the former, there being a series of wheels placed side by side on the same shaft. In the sketch, *A B C* are the different chambers in which the wheels rotate; ***a a*** are the guides for directing the steam against the wheel buckets and *b b* are the wheel buckets. One drawing in the patent specifications of Last has a very modern appearance, in that he shows a compound turbine built up of two parts, the high pressure and the low pressure, in each of which is a series of compound wheels. The high-pressure and low-pressure sections are connected by a pipe, and their arrangement resembles that of some of the turbines built to-day.

Parsons, 1885.—With a patent issued in several countries in this year, the Hon. C. A. Parsons, who was the first to place the turbine

on a commercial basis, enters the field. In all his work he has adhered to the reaction turbine and is responsible for the successful development of the compound reaction motor. In Fig. 24 is a section of one of his patent drawings. Steam enters at the center *A*, and passes right and left between the series of guide vanes at-

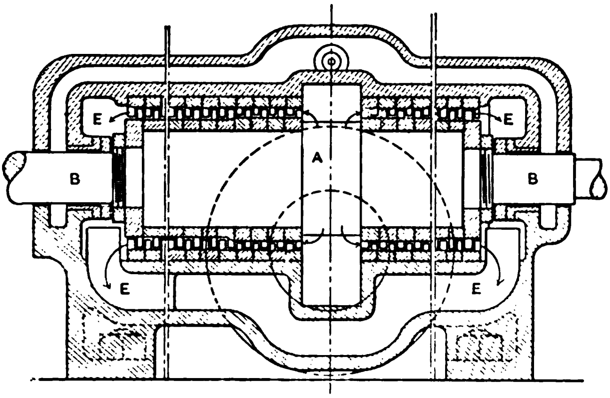


Fig. 24. Parson's First Important Patent.

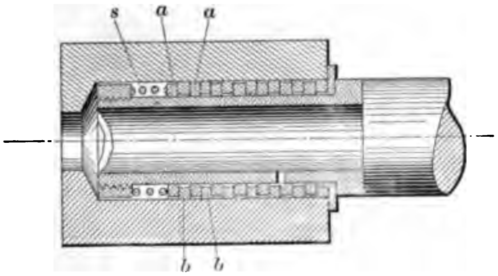


Fig. 25. Bearing for Parson's Turbine.

tached to the outer casing and the rotating blades attached to the inner drum, which has the journals, *B B*. Steam escapes to the exhaust passages *E E* and finally to the exhaust pipe shown by dotted lines. The following paragraphs are extracts from his description:

"I arrange the portions of the motor to form an approximately cylindrical figure, the whole being mounted upon the same shaft,

the first delivering into the second, the second into the third and so on. Each portion comprises a set of fixed and a set of moving vanes, the direction of motion of the actuating fluid being generally parallel, or approximately so. To balance the end pressure upon the cylinder, I mount two similar sets of rotary parts upon one shaft, one set being so placed at each side of the inlet for the actuating fluid that the entering stream divides right and left, and the exhaust takes place at both ends.

"As the speed of the motor will be necessarily high, and perfect balancing of the moving parts would not be practicable, I give to the bearings a certain very small amount of elasticity or play combined with a frictional resistance to their motion."

This refers to the well-known Parsons construction shown in Fig. 25, in which there is an annular space, between the shell of the bearing and the pocket for the shell bored out in the frame, filled with a series of metal rings. Every other ring is bored to fit the outside of the shell, but its outside diameter is smaller than the bore of the pocket, as in *a a*. The alternate rings *b b* are turned to fit the pocket, but are bored larger than the outside of the shell. The rings are forced together by a spring *s*, so that they offer considerable resistance to any lateral movement of the bearing.

He says: "The lubrication is effected by forcing lubricant through pipes to the parts to be lubricated and for this purpose a pump can be employed. To prevent leakage past the shaft at the end covers of the casing, which, when steam is the actuating fluid, would be inconvenient, I form annular recesses in the covers

aft ends, and place these recesses in communication

which a partial vacuum is maintained by suitable

a steam jet."

ar, 1888, Parsons took out a patent in which the

re arranged in groups, each successive group being

ter than the preceding one, to allow the steam, as

o flow through larger spaces, as required by the in-

e specific volume of the steam. He also proposes to

n-tight joints at the bearings by admitting water under

o an annular groove passing around the bearing. He

utting a spiral groove on the shaft, at the section where

ir groove occurs, with the idea that when the shaft is

revolving at high speed, the spiral will diminish the quantity of water forced into the turbine casing by the air pressure, when the turbine is running condensing.

Altham, 1892.—A compound turbine consisting of two rotating wheels, one inside of and concentric with the other, is the invention of George J. Altham. The buckets of the inner wheel are arranged in its outer periphery and those of the outer wheel in its inner periphery, so that steam will act alternately on the inner and outer

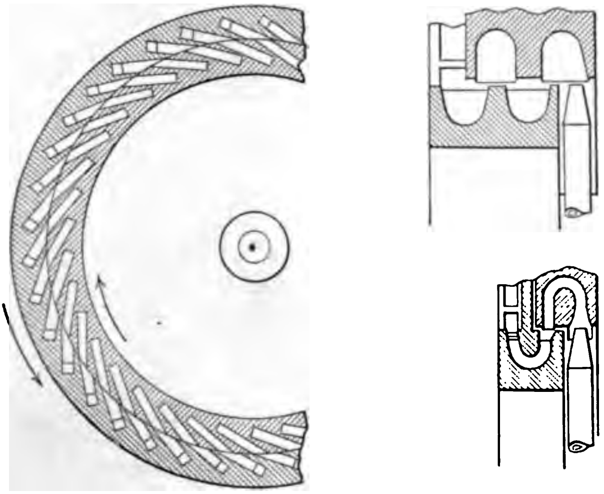


Fig. 26. Altham Two-wheel Turbine.

wheel, and successively on the different buckets of both wheels, the arrangement being such that the wheels rotate simultaneously in opposite directions. Fig. 26 shows at the left a cross-section of the rims of the two wheels in which the buckets are cut. The other sketches show longitudinal sections of the rims. Steam is discharged from the nozzle into one row of buckets of the outer wheel, whence it passes to the first row of buckets of the inner wheel, thence to the second row of the outer wheel and finally to the last row of the inner wheel, from which it discharges into the turbine casing. In the smallest sketch, Fig. 26, the construction is indicated where there is only a single row of buckets in each wheel. In this same year patents were granted to J. F. McElroy

for a turbine with U-shaped channels, but with one set of vanes attached to the casing. See, also, Imray's patent of 1881.

Dow, 1893.—When turbines first began to come into prominence in this country, the one invented by J. H. Dow was one of the three or four that were most frequently mentioned. His first patent was issued in 1887, and later several others were taken out, but the one showing the most completely worked out design was issued in 1893. All of the Dow turbines are of the radial outward-flow type, consisting of alternating rings of rotating and stationary vanes, and in this respect resemble one of Wilson's inventions of

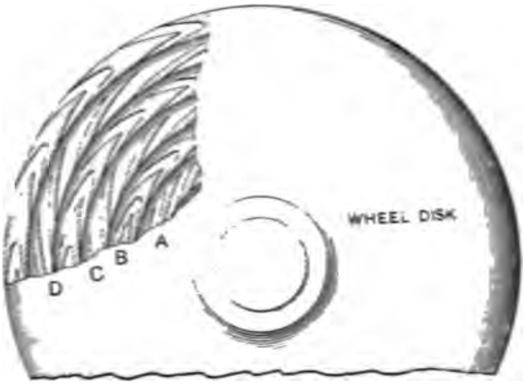


Fig. 27. Dow's Patent of 1893.

1848. In Fig. 27, taken from his latest patent, *A* is the ring of stationary vanes directing the steam against the ring of rotating blades *B*, and *C* is another ring of stationary vanes, *D* a ring of rotating blades, etc. A peculiarity of the drawing shown is that the stationary vanes are not curved at their inlet ends in a way to guide the steam into them in the direction in which it leaves the rotating blades, except as the latter might be designed so that steam would leave them in the direction in which the wheel is turning, which would be an inefficient arrangement. As actually constructed, however, the guide vanes were curved correctly and the turbine was built along the lines shown in the reference to it in Thurston's manual of the steam engine. In the patent of 1887 there is a single shaft on which are two disks facing each other, having annular rows of vanes cut on their inner faces. Between

these two disks is a central stationary disk with annular rows of guide vanes cut on each of its faces. The arrangement is shown in Fig. 28. Steam enters at the center, and flows radially outward between the vanes on each side of the central disk. In his latest patent Dow compounds his turbine still further by providing several rotating and stationary disks ranged along the shaft on each side of the center. Steam enters at the center and gradually works outward toward both ends of the turbine.

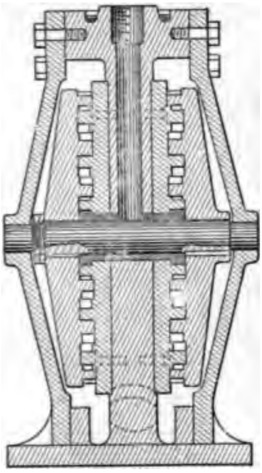


Fig. 28. Dow Turbine with Two Disks.

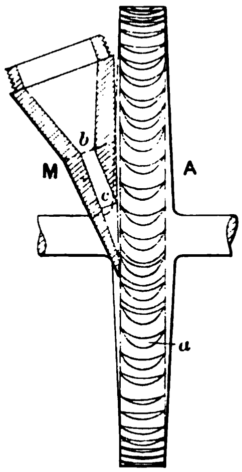


Fig. 29. De Laval Turbine with Diverging Nozzle.

De Laval, 1894.—De Laval's most important patent relates to his expanding nozzle, in combination with a turbine wheel. It is interesting to note in this connection that the expanding nozzle was patented in this country in 1867, patent 64,539, for steam injectors. De Laval, however, was the first to apply the principle of a diverging nozzle for the expansion of steam to a turbine. The two broadest claims of the patent are the following:

1. The combination with a bucket or turbine wheel, of a stationary nozzle opening adjacent to the wheel and having its bore diverging or increasing in area of cross section toward its discharge end, whereby the elastic fluid under pressure is expanded in passing through the diverging nozzle and its pressure is converted into velocity before the jet is delivered against the wheel.

2. The combination with a bucket or turbine wheel, of a stationary nozzle opening adjacent to the wheel and provided with a contracted receiving portion and with a discharge portion having its bore diverging or increasing in area or cross section toward its discharge end.

Maison Breguet, 1894.—Judging from the illustration accompanying this patent, it introduces no new principle that was not included in the invention of Hartman's compound impulse turbine, the patent for which was taken out in 1858. That is to say, the

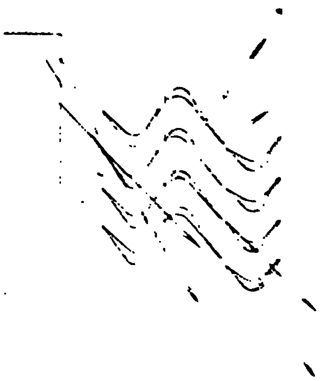


Fig. 30. From Patent issued to the firm of Breguet, Paris.

illustration shows a converging nozzle in connection with rotating
of blades alternating with stationary vanes. This is nothing
than what is shown in Hartman's patent drawings.
ly the text of the Breguet patent reads, however, the
rently had in mind the improvement of the De Laval
if such is the case he evidently intended to imply the
erging nozzle instead of a converging nozzle in con-
th a compound turbine. Putting this interpretation
atent it is of importance as the first to be issued upon this
on of elements preceding as it does, the Curtis patent
ntroduced the same principle by about two years. The
m of the invention states that in the De Laval turbine
h a circumferential velocity of the turbine of 420 meters,

if the steam has a velocity of 1,100 meters, it still discharges from this turbine with a velocity of 440 meters, and this velocity is much higher when the circumferential velocity of the turbine is less. The idea that has naturally come to us is to utilize anew this lost velocity in a second turbine mounted on the same axis, and even in exceptional cases in a third, so as to increase the use of the turbine. We affirm as our property the invention of the compound steam or gas turbine, in which the steam, or gas, after having lost a part of its live force in the turbine buckets, finally loses the remainder in the buckets of one or of several other disks mounted on the same arbor."

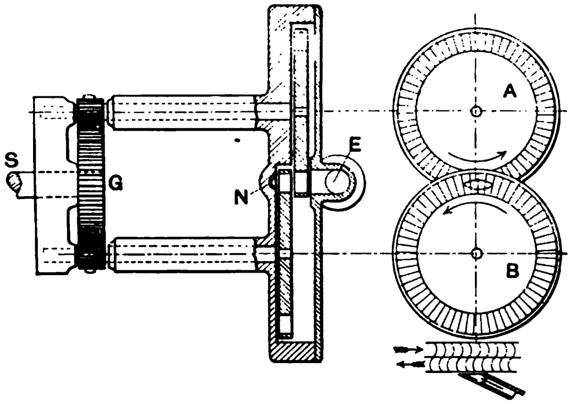


Fig. 31. Seger's First Patent.

Seger, 1894.—Seger's turbine has been built and used to some extent abroad. His first patent specification, issued in 1893, shows an arrangement of wheels indicated in Fig. 31. Like Pilbrow and B. T. Babbitt, he seeks to secure a moderate speed of rotation by using two wheels turning in opposite directions. The steam, in leaving the first wheel, impinges directly against the second without any intervening guide vanes. The turbine wheels are on separate, parallel shafts, and at *G* are the gears by which the motion of the wheels is transmitted to the driving shaft, *S*. *N* is the nozzle through which steam enters, and *E* the exhaust passage. His claim is for "a steam turbine in which the turbine wheels are placed in close proximity to each other, and are com-

bined with one or more steam conduits discharging into the sides of said wheels in such a manner that the steam passes through the wheels in the direction of their axes, and in which the shafts are arranged out of line with each other so that the wheels only partly overlap each other."

In his patent of 1894, Seger shows wheels arranged on the same axis, but rotating in opposite directions. A feature of the patent is the method for fastening the buckets in diagonal slots cut in the

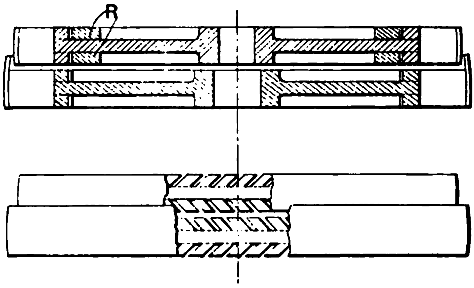


Fig. 32. Arrangement of Wheels.

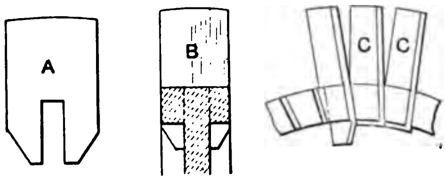


Fig. 33. Vanes of Seger Turbine.

wheel rims. The lower view, Fig. 32, shows these slots, and in the upper view the rings, *R*, forced inside the rim, hold the projecting ends of the buckets in position. This construction will be evident from Fig. 33, where *A* is one of the buckets. At *B* the bucket is placed in the rim and at *CC* its projecting ends are bent over underneath the rim. In 1897 Seger issued an English patent upon an arrangement of his turbine by which the belted connection could be used for driving the low-speed shaft, Fig. 34, from which power is taken. Here *A* and *B* are the turbine wheels rotating in opposite directions and attached to the ends of shafts which carry, at their outer ends, small belt pulleys. On the shaft, *S*, are two pulleys, W_1, W_2, of equal diameter, one of which is fast to the shaft,

he other is loose on the same shaft. The belt passing around ,everal pulleys, as indicated, transmits power from the turheels to the main shaft.

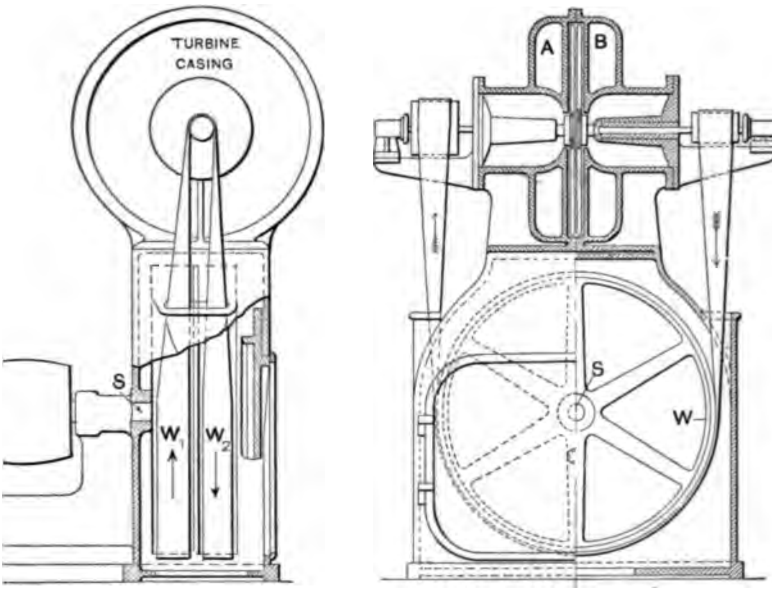


Fig. 34. Front and Side Elevation of Seger Turbine.

cElroy, 1894.—This year marks the beginning of the imnt patents on turbine wheels of the Pelton type for use with astic fluid. Two patents were issued, one to J. F. McElroy s country, and one in England to Professor A. Rateau. The gement of the nozzles and wheel of McElroy's invention is n at *A* in the sectional view, Fig. 35. Nozzles with diverging hpieces are used. At *B* is a section of a bucket in a plane lel with the shaft, together with an enlarged view of one of the es. At *C* is a section through a bucket taken in the other tion. While an efficient type of bucket is shown, and Mc-/'s claims as to its shape are broad, taken by themselves, they ombined with certain other constructive features which limit cope of the patent. What he claims is first a wheel comprising id metal ring having a series of inclined pockets therein, the

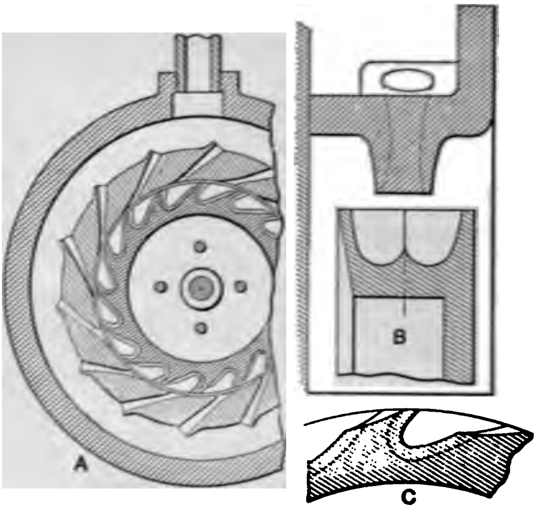


Fig. 35. An Early "Pelton" Type.

pockets of each pair being divided by a tapering ridge in combination with a circular steam ring having a circular series of nozzles; and secondly, pockets as above described, but with a flat inclined cut-away portion for each, as shown in the sketch.

Rateau, 1894.—Professor Rateau, of Paris, was one of the earliest to experiment with a steam turbine having a single wheel of the Pelton type. The essential features of his English patent on this are shown in Figs. 36 and 37, which represent the wheel vanes. He intended primarily to produce a reversible wheel and uses buckets projecting radially, with double cancave surfaces, *A, B,*

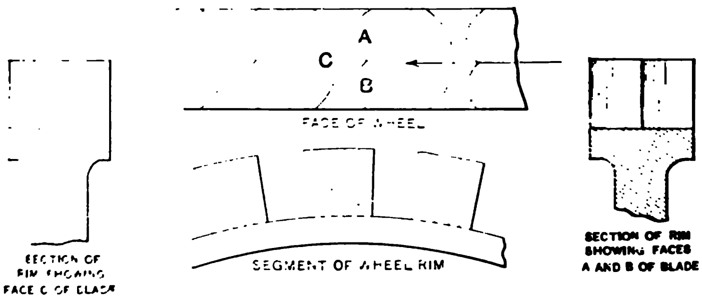


Fig. 36. Rateau's Reversing "Pelton" Wheel.

Fig. 39, which form a dividing wedge at the center, just as in the Pelton water wheel. For reversing the direction of rotation, he uses steam jets flowing in the opposite direction and impinging against the backs of the blades. The backs are shaped as shown at *C,* with a single concave surface, instead of with the double curve, in order to avoid any obstruction to the steam when running in the normal direction. While the single curve is less efficient than the double curve, it answers the requirements for the brief periods during which the turbine has to be reversed.

When the wheel is to be designed for forward motion only, Rateau presents the construction of Fig. 37. *A* and *B* are two

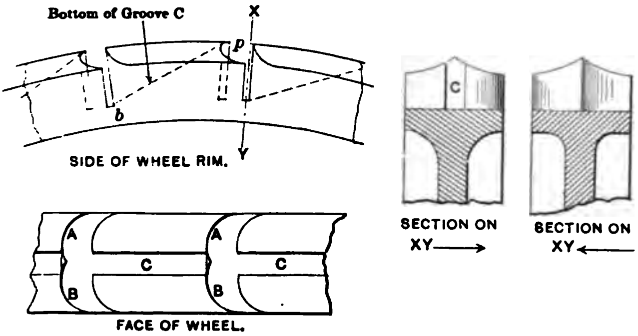


Fig. 37. Wheel for Forward Motion Only.

concave vane surfaces and at the rear of each bucket an inclined groove, *C,* is cut, represented by the dotted line, *bp,* in the upper view. This allows the jets of steam to strike the buckets, one after the other, without interference from the successive buckets as they come into position. The sectional views at the right are taken on the line, *XY,* looking, in each case, in the direction of the arrow drawn under each view.

While it is not introduced as a definite claim in this patent, Professor Rateau mentions that where the speed of the fluid is too great it may be necessary to arrange these turbines in series on the same or independent shafts, in which case the openings of the nozzles should all be designed to deliver the same relative quantity of fluid at the same moment. This he would accomplish by using distributing valves for supplying steam to the nozzles, and having

these valves all operated positively from the same source so that they would act in unison.

Parsons, 1895.—In the method of governing used on Parsons turbines, an oscillatory motion is given to the throttle valve by an eccentric driven by the turbine, and the extent of this movement is controlled by a governor. In the illustration, *A* is a double-seated throttle attached to a valve stem, *B,* which is connected with a piston, *C,* working in a cylinder above the valve chamber. At *D*

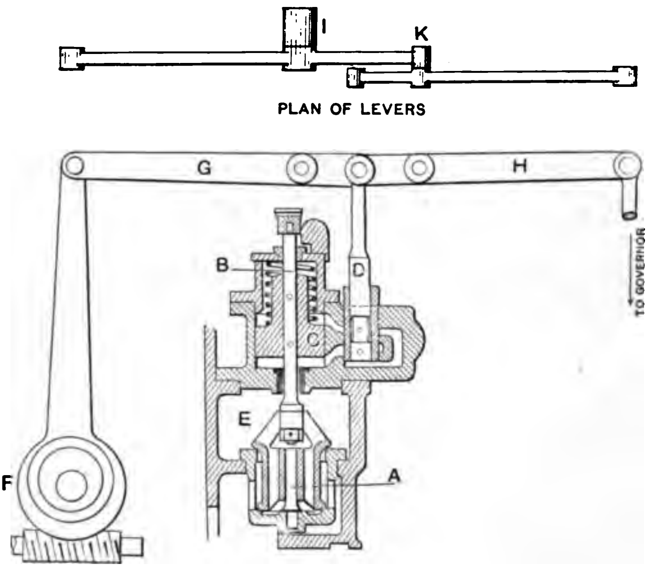


Fig. 38. Parson's Governing Arrangement.

is a pilot valve for controlling the motion of the piston. The steam enters the chamber, *E,* and flows downward through the valve to the turbine. An opening from *E* to the space below the piston allows the steam to push the piston upward against a spiral spring which pushes it downward in case the steam pressure underneath the piston is relieved. When pilot valve *D* closes the port leading from the space below the piston, the pressure maintained under the piston causes the latter to rise and with it the valve *A,* but when valve *D* uncovers the port, steam escapes from under the piston and

passes around to the top, and together with the spring serves to close the valve.

A floating lever mechanism is used for controlling the pilot valve. At *F* is an eccentric driven by a worm and worm-wheel, which oscillates lever *G* about its fulcrum, *I*. At point *K*, on lever *G*, the lever *H* is fulcrumed. One end of lever *H* is connected to the governor and the other end to the pilot valve *D*. The pilot valve, therefore, is controlled both by the motion of the eccentric and the motion of the governor. The eccentric keeps the pilot valve and hence the main throttle valve in constant oscillation, while the movement of the governor changes the positions of the limits of this motion. For example, if the turbine were running with a light load, the valve would oscillate in the lower end of its possible path of travel and would shut off steam entirely at each oscillation; but if the turbines were heavily loaded, the valve would be moved upward by the governor and its path of travel would be located higher, so that steam would flow through the valve continuously, although it would be throttled more or less as the valve moved up and down under the action of the eccentric.

Sebastian Z. de Ferranti, 1895.—The patent taken out by this inventor is to be classed with Hartman's patent of 1858 and that of the Société Anonyme Maison Breguet, 1894, all three of which propose a compound turbine containing certain features employed in the Curtis patents now used by the General Electric Company. It is to be noted, however, that, like his predecessors, Ferranti fails to specifically state that he wishes to employ a diverging nozzle in combination with a compound turbine, which is an important feature of the Curtis type of wheel, although he says that he intends to utilize fluid in the wheel, "after complete expansion and the acquisition of the maximum velocity," which, under certain conditions, can only be attained in a diverging nozzle. He advocates the use of superheated steam and also refers to gas turbines. The following is an extract from his specifications:

"I construct impact engines in which the working fluid impinges after complete expansion and acquisition of the maximum velocity, upon semi-circular rotating blades fixed round the rim of a motor wheel. The working fluid enters the blades at a high velocity and has its direction reversed, a portion of its energy being turned into

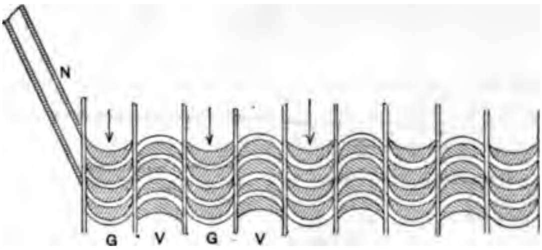


Fig. 39. One Plan for Compounding.

work, and rotates the wheel, and then leaves at the other side of the blades at a diminished, though still high velocity. I then pass it through a set of standing semi-circular blades of exactly the same description as the rotating blades, but with grooves in the opposite direction, which reverses its direction, bringing it back to the original direction of motion, when it strikes the blades of the second wheel and delivers up a further portion of its energy and comes out at a reduced velocity. This process is repeated until the steam issues from the last set of blades with practically no useful velocity, it having given up nearly all its energy to the rotating

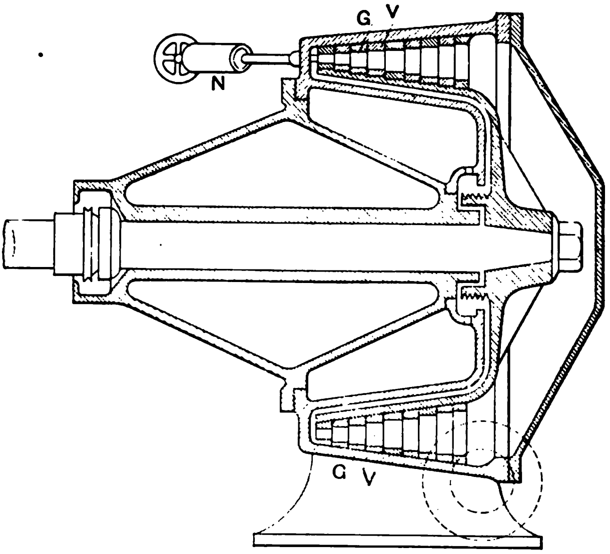


Fig. 40. Ferranti's Turbine.

blades of the wheel. . . . . The object is to convert the whole of the energy and pressure in the working fluid into velocity of the particles, which then react backward and forward through the rotating and standing blades of the machine, thus constituting an impact multiple re-active engine.

"The engine may be made with one or more expansion tubes according to the power it is desired to obtain. More or less of these expansion tubes may be used and actuated by the governor according to the power required for the time being. . . . . The expansion tubes stand tangentially from the periphery of the wheel and at a

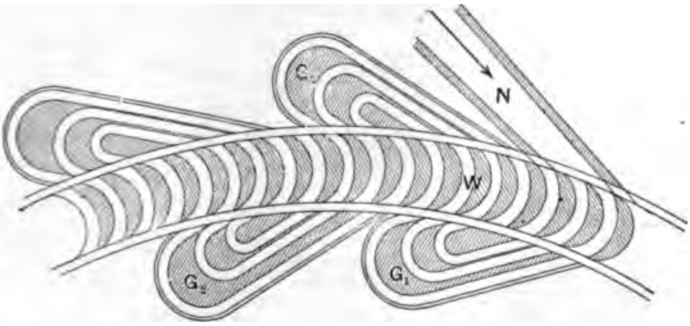


Fig. 41. A Second Plan for Compounding.

slight angle to the side of the wheel so as to deliver its working fluid in the most suitable position."

Fig. 39 shows the principle of his scheme, N being the nozzle, G a set of rotating vanes, V stationary guide vanes reversing the direction of motion, and so on. The design of turbine proposed by him is illustrated in Fig. 40, where N is a nozzle and G and V the rotating and guide vanes respectively, the rotating vanes being attached to a conical drum on the end of the turbine shaft and the guide vanes attached to a casing on the turbine. In this design he plans to have an increasing area for the steam as it flows through the turbine, after Parsons' plan, which is somewhat contrary to the statement of his specifications. In Fig. 41 is still another proposed arrangement in which the steam, directed by the nozzle, N, impinges against the wheel vanes, W, and is then taken up by the U-shaped passages, G_1, and returned to the wheel vanes, where it is again

taken up by the U-shaped passages, G_2. This is another modification of the schemes advanced by Wilson, Perrigault & Farcot and a number of other early inventors, and, as previously stated, later by Profs. Riedler and Stumpf.

Curtis, 1896.—The important group of patents taken out in this year by Mr. C. G. Curtis, inventor of the turbine manufactured by the General Electric Company, cover most of the basic principles of this turbine. The leading feature of the Curtis machine is the combination of a diverging or expanding nozzle with a compound

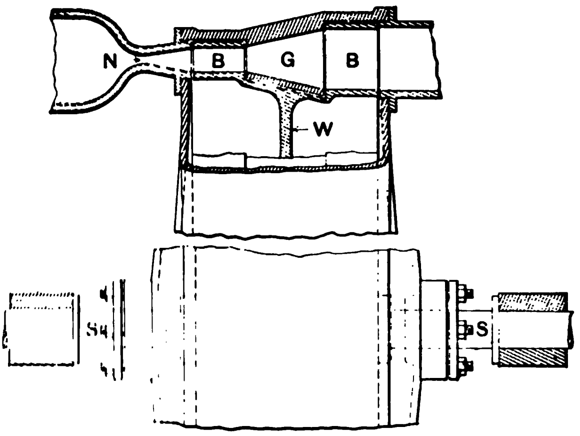


Fig. 42. Curtis Turbine.

turbine wheel, although other features are included which had been found by the inventor to be necessary to the successful operation of a turbine of this form of construction. As already stated —vious inventors have patented turbines in which were combined e for directing the steam against the blades of the rotating of a compound turbine, but the Curtis patents are the first ly claim the diverging nozzle as a part of the combination, y are, furthermore, the first ones to fully explain a practical d for carrying out the design so as to make an operative and mical machine. While the requirements for the success- peration of a compound impulse turbine were outlined by rhouse in his specifications of 1877, in which he provided for gressive expansion of the steam from inlet to exhaust, by

using passages of gradually increasing areas, his patent was limited to the type of construction having a succession of compartments, in each of which is a single turbine wheel.

In Fig. 42 are shown the elements of the Curtis turbine. The shaft, *SS*, carries a turbine wheel on which are two annular rows of buckets, *BB*. At N is the expanding nozzle for directing the steam against the first ring of buckets, after which it passes through a group of guide vanes, *G*, to the second ring of wheel buckets. In the operation of a turbine of this type the ideal condition would be attained if the expansion of the steam could be complete in the nozzle and then it were to pass through the turbine in virtue of its

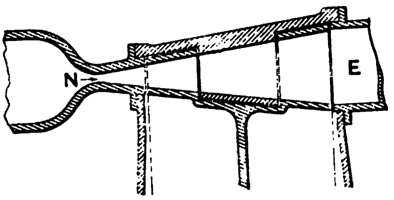


Fig. 43. Preferred Construction for Curtis Turbine.

inertia, giving up a part of its velocity to the first ring of buckets and the balance to the second or succeeding rings. This would allow the wheel to run at a comparatively low velocity, depending upon the number of times the wheel was compounded. That is, if there were a single wheel, as in the De Laval type, it should run at approximately half the velocity of the steam; but if there were two wheels instead they could be run at a lower velocity so that the steam in leaving the first wheel would have a residual velocity to be taken up by the second wheel. By carrying the compounding still further a still slower speed of rotation could be used. Steam, however, is an elastic body, easily diffused, and has so small a mass that its inertia will not carry it through a succession of vanes in the above manner, unless there is an additional propelling force generated to overcome the frictional and other resistances during the passage through the vanes. This is accomplished in Fig. 42 by reserving a part of the expansion of the fluid to take place in the guide passages, *G*, which, as shown, are made diverging for the purpose. Accordingly, after the steam leaves the first set of vanes

it receives an additional impulse in the guide passages before coming in contact with the second set of vanes. The preferred construction, however, and the one which is actually employed, is shown in Fig. 43. Here the steam is expanded in the nozzle, N, to nearly, but not quite the final pressure of the exhaust pipe, E. The balance of the expansion occurs during the passage between both the rotating and stationary vanes, and the pressure within these passages is, therefore, slightly in excess of the pressure within the chamber in which the wheel is rotating.

The illustration, Fig. 44, is from the so-called "stage patent"

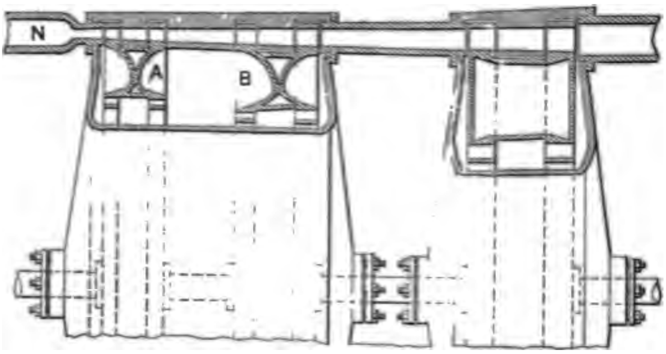


Fig. 44. Curtis "Stage" Turbine.

upon the Curtis turbine. The cut shows each stage to be composed of one or more of the compound elements that go to make up the turbine represented in Figs. 42 and 43. In each of the two stages the wheel-and-bucket arrangement differs one from the other. In the first casing are wheels A and B, each carrying two sets of rotating rings or vanes, and in the second casing is a single wheel with two sets of blades. The advantage of dividing the turbine into stages in this way, is that there is less leakage between the guide vanes and the wheel vanes, since the differences of pressure are less; and there is also less diffusion of the steam since the number of rows of vanes for the steam to pass through in each stage is less than would be the case if all the rows were combined together in one casing and the steam were compelled to pass through them in virtue of the velocity acquired in the nozzle at the beginning.

A third patent taken out in this year deals with the problem of governing, and Mr. Curtis shows methods for changing the quantity of steam supplied to the turbine without throttling the pressure or reducing the velocity of flow. Obviously an expanding nozzle of certain proportions is adapted only to the steam pressure for

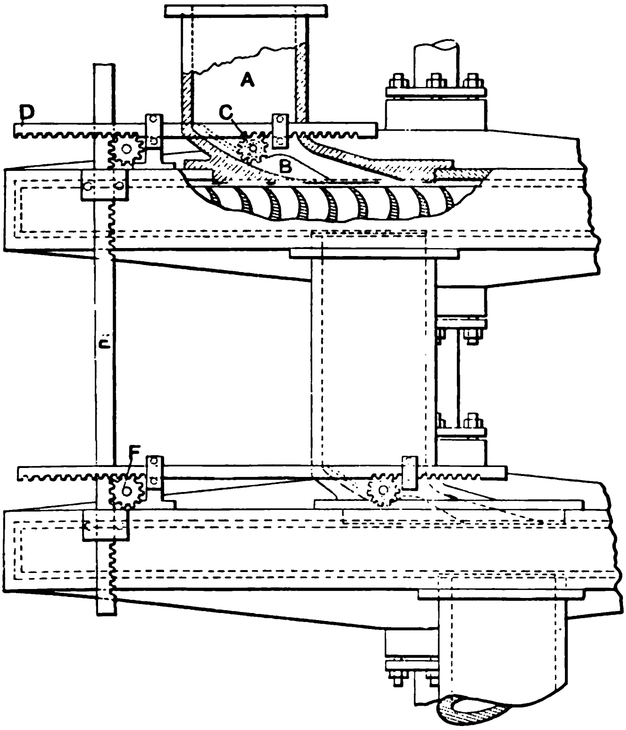


Fig. 45. Curtis' Plan for Governing a Compound Turbine.

which it was designed, and when the pressure is throttled the nozzle does not operate at its highest efficiency. It is proposed by Mr. Curtis to avoid this by using a nozzle of rectangular cross-section with one side adjustable in or out, regulating the quantity of steam flowing through the nozzle without making a great change in the ratio of the inlet and outlet areas. In Fig. 45 is a diagram showing the principle proposed, where the turbine is divided into two or more stages. *A* is the steam inlet, terminating in a nozzle

having a sliding piece, *B*, operated by the rack, *D*, and pinion, *C*. This rack gears with another pinion, which transmits motion through the rack, *E*, to the pinion, *F*, and this in turn operates a similar sliding piece in the nozzle, directing steam against the second nozzle. The claims for the apparatus as used with a compound turbine cover first the principle of governing by changing the volume without great variations in the velocity of the steam

Fig. 47. [illegible]

as equivalent [illegible] the simultaneous proportionate [illegible] passages leading to turbine or [illegible] the turbine.

Kollmann, [illegible] turbine with steam a fluid [illegible] of the it impinging [illegible] been many attempts to [illegible] referring to mix a heavier gas [illegible] or some less volatile [illegible] similar

to the way steam and water are mixed in a steam injector. One of the earliest attempts to do this was by Pelletan in 1838. Others who have proposed fairly good arrangements of this character are Millward in 1866, Crumlisk in 1869, Miller and Collins in 1896, and Lundell a year after Bollmann, in 1898.

The chief interest in Bollman's invention centers in a new type of expansion nozzle rather than in the plan for using a heavier working fluid. The nozzle consists of an annular slot between two disks *B* and *C*,—the former of which is adjustable. Steam enters through the pipe *A* and flows radially out through the annular slot between the disks. Inasmuch as the diameter of this slot is small where the steam enters and is larger where it leaves, the steam expands in flowing through the slot, although the faces of the disks do not diverge. Escaping from the nozzle, the steam enters the space *D D* and there combines with air and passes radially outward to the guide vanes *E E E* and the wheel vanes *F F F*. The plan is to use this combined fluid in a turbine working on substantially the principle of the Curtis turbine.

With this invention the review of steam turbine patents will close. The patents chosen for these pages, while representing only a part of the best work of inventors of the past century in perfecting the steam turbine, point the way by which success has finally been attained and indicate the directions that the paths of progress in this field will most likely take in the future. The author believes that the hints contained in the descriptions of these inventions will prove of real value to inventors who are at work on the steam turbine problem, as they will give at least a limited idea of what has already been accomplished and will enable inventors to work more intelligently. In subsequent pages reference will be made to some of the later inventions in connection with descriptions of the leading turbines now on the market.

A WORD WITH INVENTORS.

In reviewing the patents upon this subject, a great many more features have been found in the specifications which would contribute to an *unsuccessful* turbine than to a *successful* one. It would be out of the question to point out all of these, but a few

of them have appeared so frequently and have been re-invented so many times at the expense of the inventor and to the profit of the patent lawyer and the United States Patent Office that it will be well to give attention to them.

[illegible] — One of these is the scheme proposed in the [illegible] patent of [illegible]. It is improbable that any turbine in which the velocity of the steam is reduced by combining with some other fluid [illegible] of the injector can prove at all [illegible]. The reason for this is that in com[illegible] proposed the kinetic energy [illegible] the vanes of the wheels [illegible]. This can be explained as [illegible]

[illegible] to combine with some [illegible] steam injector, where the steam [illegible] a certain velocity to the [illegible] and in calculating the [illegible] of impact, that

[illegible] after combining.

[illegible] of gravity.

[illegible] the velocity of [illegible]

Let [illegible]

[illegible]

W = [illegible] combined with the steam.

Then [illegible]

Now, if we suppose all the kinetic energy of the jet be used by the turbine, the capacity of the jet for doing work is represented by

$$\frac{V^2}{2g}$$

in the case of the steam, and by

$$\frac{(W+1)\ V_1^2}{2g}$$

in the case of the combined fluid. Let us assume that one pound of water or other fluid is used for each pound of steam. It is evident from above formula (1) that the velocity of the combined jet will then be one half the velocity of the steam, while the weight will be double.

The kinetic energy of the steam,

$$\frac{V^2}{2g}$$

will therefore be two times that of the combined jet, which is

$$\frac{(W+1)\ V_1^2}{2g}$$

This does not mean that energy has been lost. In this process the heat energy of the steam is first converted into kinetic energy, giving the steam jet high velocity of flow at the start; second, part of the kinetic energy is converted back again into potential energy in the form of heat or pressure, or both, after the two fluids combine; and the kinetic energy remaining is all that is available for doing work. If the turbine is to be efficient, this potential energy must be utilized by converting it again into kinetic energy, and it must be acknowledged that so many transformations would entail serious losses, if, in fact, it were possible to make them at all.

A Misconception of Reaction.—It has been demonstrated many times that the reaction of a jet of water or steam is not altered by holding an obstruction in the pathway of the jet, unless the obstruction is placed near enough to the mouth of the nozzle to choke the flow. This has not been realized by some inventors who have schemed on turbines similar to Fig. 47, where steam enters the

hollow radial arms through the trunnion *A* and discharges at the orifices *B* and *C*. Those inventors who have provided notches such as *N N* for the steam to strike against have not improved the efficiency of the machine in any way. Likewise those who have arranged for two rotating elements, one of which may be represented by the arms in Fig. 47, and the other to consist of a ring rotating in the opposite direction and containing blades *N N*, have done nothing to improve the efficiency. If the blades *N N* are properly shaped, as, for instance, in the Seger turbine, speed re-

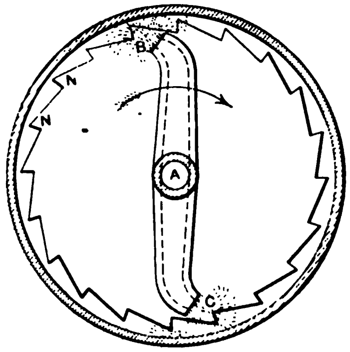


Fig. 47. A Useless Construction.

duction may be secured by this means, but there would theoretically be no improvement in the efficiency. Other inventors have attempted to produce turbines, combining the principle of the rotary engine, in which the blades move through closed compartments and the steam, after impinging against the blades, reacts against an abutment. None of these various schemes are likely to be successful, and inventors are advised to adhere to the plan of first providing means for converting as much of the potential or heat energy of the steam into kinetic energy as possible, and then using this energy to the best possible advantage according to the well-proven laws of the hydraulic turbine.

CHAPTER III

SIMPLE IMPULSE TURBINES.

The De Laval Steam Turbine.

History and Characteristic Features.—The De Laval turbine is the invention of Gustaf De Laval, the famous Swedish scientist. De Laval received an advanced technical education as a preparation for his career and has shown himself a versatile inventor along several lines in his chosen profession. Previous to his turbine

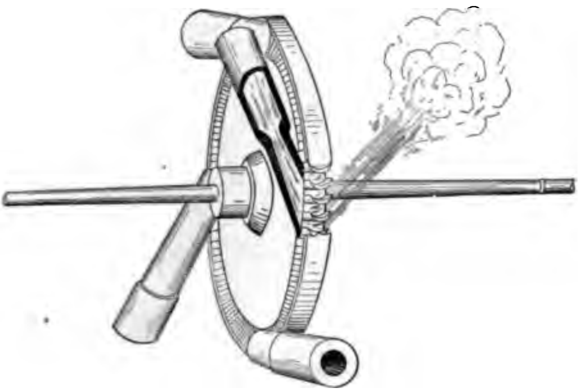


Fig. 1. De Laval Wheel, Flexible Shaft and Nozzles.

achievements his most noteworthy invention was in connection with the centrifugal separators, which have become of enormous importance in the dairy industry of the world. It was in the manufacture of these that he conceived the idea of the steam turbine as the ideal motor for the separator, which must itself run at extremely high speeds. As recorded in Chapter II., he used the reaction type at first. In 1888 he experimented with the diverging expansion nozzle, which is mainly responsible for the efficiency of the De Laval turbine. The high velocity of the steam in the diverging nozzle necessitated considerably higher wheel velocities than he had used before, and it was found to be very difficult to balance a wheel accurately enough to avoid destructive pressure

upon the bearings under such conditions. He therefore adopted the flexible shaft, which, with the diverging nozzle, is employed in the De Laval turbine of to-day. These characteristic features are represented in the familiar illustration, Fig. 1.

General Description.—Fig. 2 is an external view of an electric generating set consisting of a De Laval turbine direct-connected to double, direct-current, Bullock generators. The generator is at the left, the turbine at the extreme right, and between the two are the casings inclosing the speed reduction gears connecting the turbine and generator shafts. The turbine wheel rotates within a steel casing and on one end of its shaft is a small, double, spiral

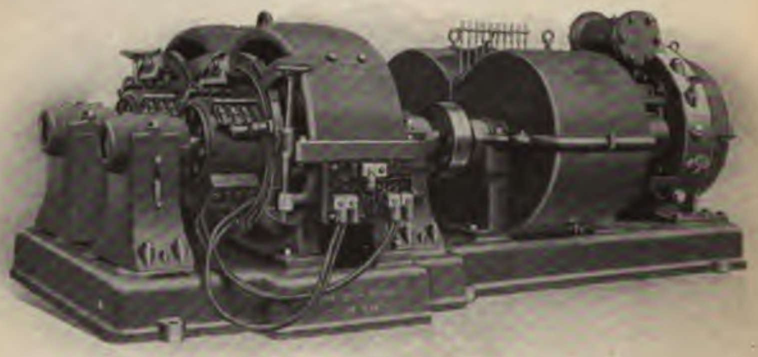
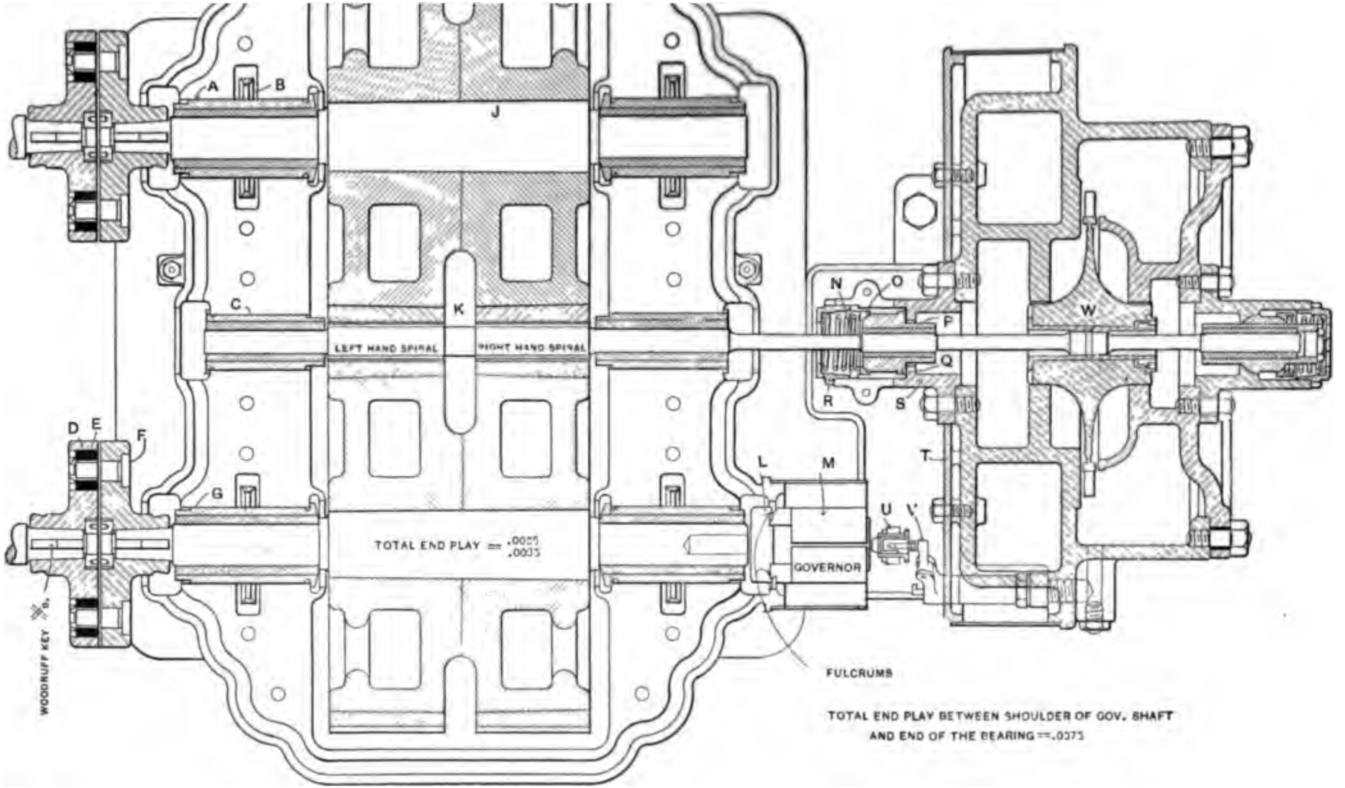


Fig. 2. De Laval Electric Generating Set.

pinion, which, in the smaller sizes, meshes with a large, double spiral gear. In the large sizes two double gears are placed, one on each side of the pinion, which thus balances the thrust of the transmission.

In the large engraving, Fig. 3, is a horizontal sectional view of a turbine taken in the plane of the turbine and gear shafts. Starting at the right, W is the turbine wheel attached to the flexible shaft, which latter is supported on each side of the wheel by bearings held in the casing by ball and socket joints. The pressure within the turbine casing is practically atmospheric pressure when running non-condensing, and is equal to the pressure of the condenser when running condensing. Under the latter conditions, these bearings



NOTE: MAXIMUM SPACE BETWEEN COUPLINGS 1/8
END PLAY ON CONNECTED SHAFTS 1/16
MAKING MINIMUM SPACE BETWEEN COUPLINGS 1/16

Fig. 3. Longitudinal Section of De Laval Turbine and Reducing Gearing.

should be tight to prevent leakage of air into the casing, and they must at the same time be able to move slightly, in case of flexure of the shaft. They are, therefore, held to their seats by spiral springs *N* bearing against a collar *O* made in the form of a socket. At the other end of the flexible shaft are the spiral pinions *K*, supported on each side by bearings *C* in the wheel casing. These pinions mesh with the gears *I I*, as indicated.

The speed reduction between the pinion and gears is about in the ratio of 10 to 1 for all sizes of turbines. The speeds of the turbine wheels range from about 30,000 revolutions per minute for a 7-horse-power to 10,600 for a 300-horse-power turbine; and the speeds of the large gears range from about 900 to 3,000 revolutions per minute. The peripheral speed of the turbine wheels ranges from about 515 to 1,380 feet per second, while the peripheral speed of the gears is 100 feet per second or slightly more, for all sizes. These speeds of the gear shaft are found to be well adapted to driving generators and other apparatus, such as centrifugal pumps, blowers, etc. Such apparatus is driven through flexible couplings taking power from the outer ends of the gear shafts. The couplings have a series of pins *F*, Fig. 3, securely driven into holes in the circumference of the driving disks, and on their outer ends have rubber bushings *E*, which fit in corresponding holes in the disk attached to the shaft belonging to the generator or other apparatus. These bushings are fitted with an internal steel bushing *D*, which slips over the end of pin *F*, to protect the rubber. This brings the wear on the outside of the rubber bushing, which presents a greater area than the inside.

The governor, shown at *M*, is of compact design and is carried by a short shaft made a taper to fit in the end of one of the gear shafts. The governor controls a throttle valve and also, in case of extreme increase in speed, opens a valve admitting air to the wheel casing by means of the lever *V*. The friction of the wheel rotating in the air checks its speed.

Nozzles and Steam Chest.—In considering the individual parts of the De Laval turbine, the first to be noted are the nozzles which direct the steam against the wheel buckets. These nozzles are arranged about the circumference of the steel casting which serves as the casing for the turbine wheel. The inner end of this casting

has an annular closed space, separate from the wheel chamber, which serves as a steam chest for the turbine, as indicated in Fig. 3. The inner ends of the nozzles open into this steam chest, as in the sectional view, Fig. 4. Here *A* is the steam chest; *B,* the nozzle; *D,* the turbine wheel, and *C,* the valve for admitting steam to the nozzle. The divergence of the nozzles depends upon the steam pressure to be used and also upon whether the turbine is to run condensing or non-condensing. If the latter, the turbine is generally fitted with both condensing and non-condensing nozzles, so that in the event of difficulty with the vacuum the machine can be operated non-condensing with a greater degree of economy. The nozzles are turned to gauge on their outside and reamed to the

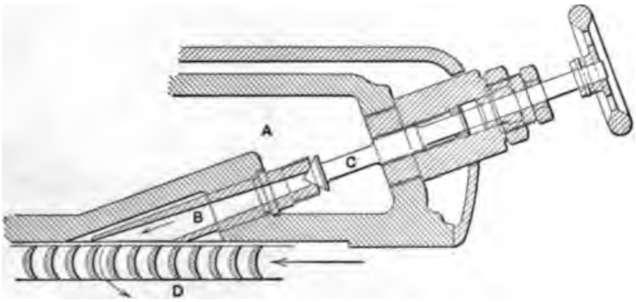


Fig. 4. Nozzle and Valve.

required taper on the inside. Over 600 reamers of different tapers are kept in the tool room of the American De Laval company for this purpose. The nozzles are simply driven into place in the casing, but are threaded at their inner ends to facilitate removal by means of a jam nut. The taper of the nozzles ranges from about 6 to 12 degrees total taper, and they are located with their outlet about 1/8 inch from the wheel blades.

Turbine Wheel and Shaft.—The turbine wheels are all made in Sweden, of a special grade of high carbon steel. They are shaped according to theoretical calculations, so as to offer nearly a uniform resistance throughout to the forces acting; but they are made slightly stronger near the center. A short distance from the periphery annular grooves are turned on each side of the wheel, making this the weak section, which would be ruptured first in case of excessive rotative speed. To further guard against danger in

the case of a wheel bursting, the steel casing is made strong enough to sustain the shock due to flying segments of the wheel; and still further, the hubs of the wheel extend into circular openings in the casing (Fig. 3), in which the hubs ordinarily run without touching. But if the wheel rim should burst, what would be left of the wheel would be out of balance and would cause the hubs to bear against the casing with great force and thus slow down.

Grooves of the shape shown in Fig. 5 are drilled and milled through the turbine rim in a crosswise direction, and in these drop-forged steel buckets are fitted. This construction enables buckets to be easily renewed, as is sometimes necessary either because of wear or accident.

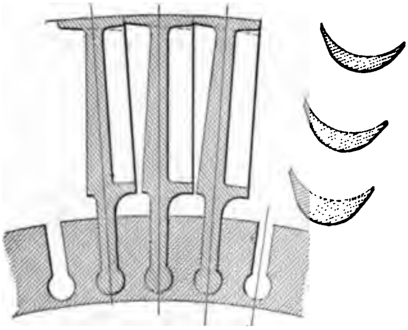


Fig. 5. Method of Inserting Blades.

In the smaller size turbines the wheels are attached to the flexible shafts by the method indicated in Fig. 3. The hubs of the wheels are bored out and a thin steel bushing is drawn into the hub by a nut at one end. The middle portion of the bushing is bored tapering and fits on a taper portion of the shaft, as indicated. This taper is the standard ½ inch per foot used by the De Laval company. After forcing the bushing on the shaft, it is pinned into place; but the wheel can easily be removed by loosening the nut and sliding it off the steel bushing. The wheels for the larger turbines are made as in Fig. 6. Here the hub is solid at the center, but each end of the hub is recessed and the flexible shaft is made with enlarged flanged ends which fit into the recesses and are bolted in place. The recesses and shaft ends are machined on a taper of ½ inch per foot.

The pinions are cut directly on an enlarged portion of the shaft, the flexibility of the shaft making an extremely accurate balance unnecessary since the wheel and shaft reach the critical speed, so-called, at about $\frac{1}{5}$ to $\frac{1}{8}$ of the normal number of revolutions of the wheel, at which point "settling" takes place and the parts proceed to rotate about their center of gravity instead of about their geometrical center.

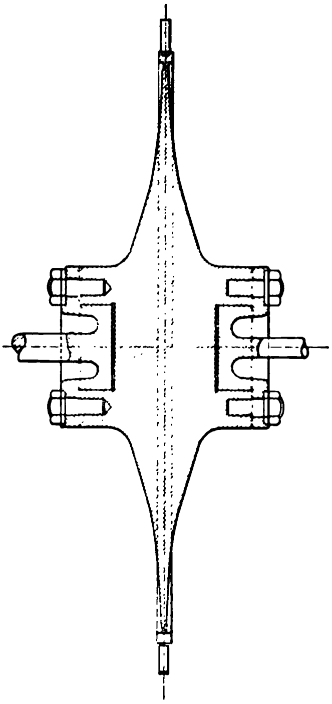


Fig. 6. Section of Turbine Wheel of the Larger Sizes.

Gears.—Next in importance to the turbine wheel, and probably first in importance in so far as the successful operation of the turbine is concerned, are the gears used to reduce the speed of the turbine shaft to a point where it is practicable to utilize the power. It was a radical step on the part of De Laval when he first attempted to run gearing at so high a speed as these gears operate, and it is safe to say that previous to the time when De Laval demonstrated that gears would run at a linear velocity of upward of 100 feet per second, it would not have been supposed possible.

The pinions are made of .60- or .70-point carbon steel and are a part of the flexible shaft. The gears are of mild .20-point carbon steel of a grade similar to that used for locomotive wheel tires. For turbines up to 30 horse-power the gears are of solid steel; but for sizes above that they are made with cast-iron centers with rims of mild steel. The teeth are of fine pitch, ranging from about .15 inch in the smallest to .26 inch in the largest sizes. The success at running these gears at high speed is due, in part, to the fine pitch and the spiral angle of the teeth, which thus brings a large number of teeth in mesh at one time, making the working pressure at each tooth very light, and re-

ducing the likelihood of abrasion. The dimensions of gears and pinions for four sizes of turbines are shown in the accompanying table:

PINIONS.

H. P.	Outside Diameter.	Number of Teeth.	Depth of Teeth.
10	1.077	21	.075
75	1.53	19	.1169
110	1.82	23	.1169
300	2.65	31	.1275

GEARS.

H. P.	Outside Diameter.	Number of Teeth.	Depth of Teeth.
10	10.1	208	.075
75	15.7	208	.1169
110	18.89	250	.1169
300	29.29	362	.12750

Oiling Arrangements.—In the high-speed bearings oiling is accomplished by having a shallow spiral groove turned in the shell, which allows the oil to reach every part of the bearing. In a 100-horse-power machine this groove is about 1/64 inch deep and 1/2-inch pitch. In connection with the oiling arrangements for the bearings, reference should be made to the design of the wheel casing in which the bearings are located. This casing is in two halves, divided on a horizontal plane, and the upper edge of the lower half has an oil groove running around it, as shown in Fig. 3, to catch any drip that may work in between the two halves. The oil is carried down into pockets in the casting, where the ring oilers reach it, and these pockets are piped to a gauge glass, to indicate the quantity of oil in them. The oiling of the various bearings is effected by means of a single sight-feed lubricator having tubes leading to them.

The Governor.—Reference has now been made to most of the principal parts of the turbine, with the exceptions of the governor and throttle valve which it controls. These are shown in Figs. 7 and 8 respectively. The governor is held in the end of one of the gear shafts by the taper plug *K*, Fig. 7, and is made cylindrical in form, with its outer shell *B B* cut longitudinally into two halves

which form the governor weights. These weights are fulcrumed at *A A* and have pins *C C* which press against a collar *D* which takes the thrust of the spiral springs located within the governor. The movement of the governor is transmitted through the center spindle *G* to the bell-crank lever *L*, which is balanced by a spiral spring *N*. The shaft supporting this lever passes through the valve casting on the inside of which are a pair of arms connecting

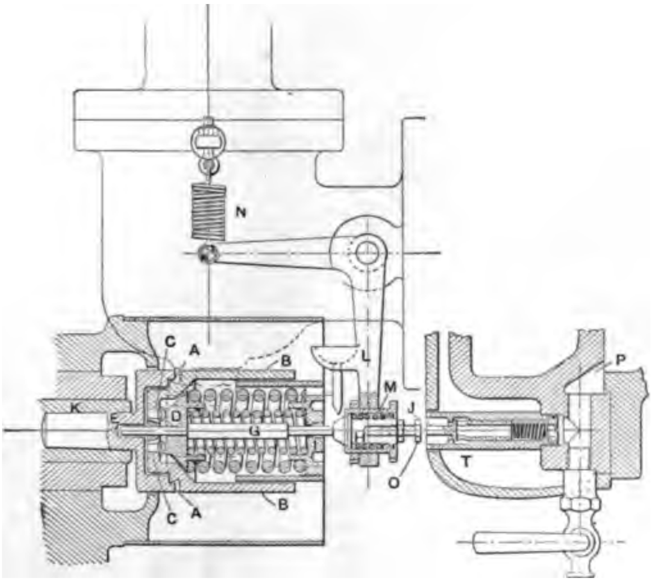


Fig. 7. Governor and Connection with Throttle Valve.

with a double-seated throttle valve as shown. In the steam pipe above the valve is a wire cylindrical screen, to prevent any large particles of scale or other material likely to damage the turbine from passing through. It is to be noted that the connection between the center spindle *G* of the governor and the bell-crank lever *L* is a flexible connection, and that at the right is a valve *T*, which connects through the passage *P* with the wheel casing. In case the throttle valve should stick and the turbine speed go up, the gov-

ernor would have power enough to overcome the pressure of the spring at the connection *H*, and the pin *O* would strike the spindle of the valve *T*, which latter would admit air to the vacuum chamber in which the wheel revolves. This would immediately put an air brake on the wheel and prevent an acceleration of speed. If for any cause the speed becomes excessive this action takes place. In a paper read before Society of Arts, Boston, in 1904, Charles

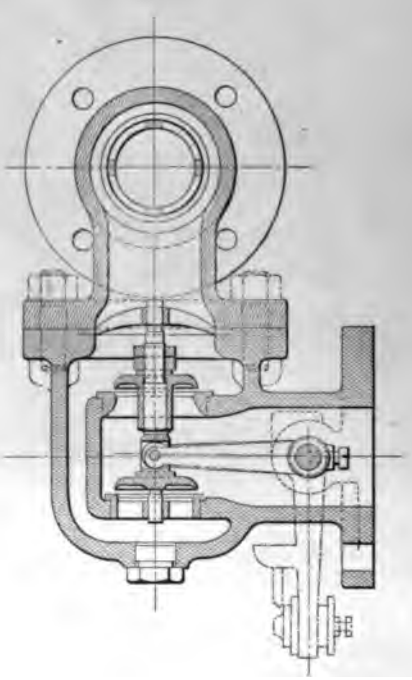


Fig. 8. Throttle Valve.

Garrison states: "To show the action of the vacuum breaker more clearly, I started a 150-horse-power turbine with all nozzles open, the nozzles being designed for 150 pounds gauge pressure and 26 inches vacuum. The condenser was shut down and the turbine exhausted against the atmosphere, and with these conditions the turbine would not come up to full speed with no load."

Special Applications of the De Laval Turbine.

The turbine, as built by the several De Laval companies, by C. A. Parsons & Co. in England, and Sautter, Harlé & Co. in France, has been applied to many special uses besides that of dynamo driving. In the United States more installations of this character have been undertaken by the De Laval company than by any other manufacturer.

Application to Centrifugal Pumps.—Until recently the centrif-

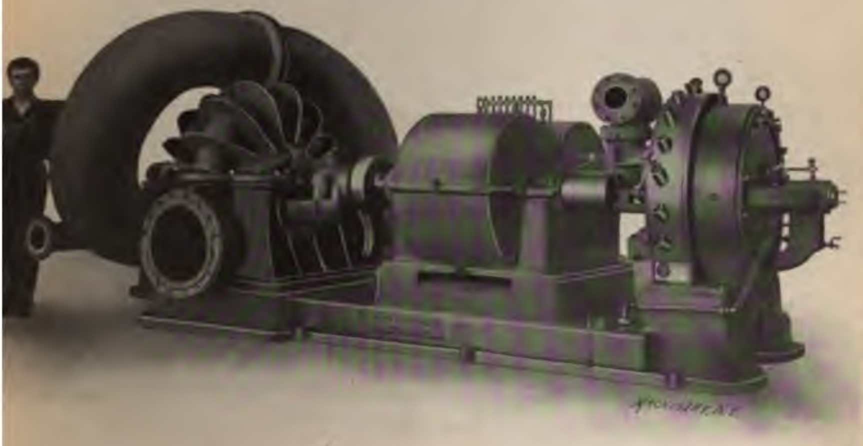


Fig. 9. Application to Compound Centrifugal Pump.

ugal pump has not been considered as efficient as the plunger pump, mainly because of the low speed and imperfect design of such apparatus; and it has also been adapted only to low lifts. As the De Laval Steam Turbine company, however, were building high-speed turbines, which ran at almost exactly the speeds required for maximum efficiency in centrifugal pumps adapted to the different sizes of turbines, it was decided to design a series of pumps, of improved mechanical construction, which would admit of the high speed necessary, and which would also enable the pumps to work against high heads. The smaller units consist of a

single pump, but the larger units have two pumps driven by the double-gear turbine, one pump being connected to each gear shaft, permitting their operation in parallel for low pressure, and in series when high lifts are desired. Standard sets are built in sizes from 7 to 300 horse-power for all heads up to 300 feet, handling from 90 to 26,000 gallons per minute. Lately a high-pressure pump has been developed with the forcing pump having a runner of very

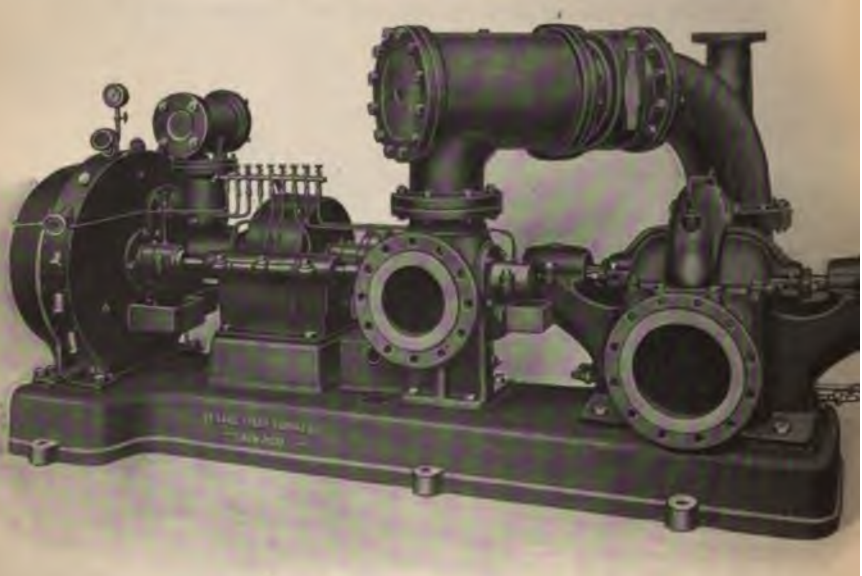


Fig. 10. High-pressure Centrifugal Pump Connected in Series.

small diameter attached to the turbine shaft and rotating at the extremely high speed of that shaft. The centrifugal force developed by the runner under these conditions makes it possible to pump against heads of 600 to 1,000 feet, and to use the pump for boiler feeding. A runner operating at such high speed, however, would force out the water more rapidly than it could draw it in by suction, or under atmospheric pressure only, and hence the pressure pump is flooded with water at sufficient pressure to ensure an adequate supply, by a pump on the geared shaft.

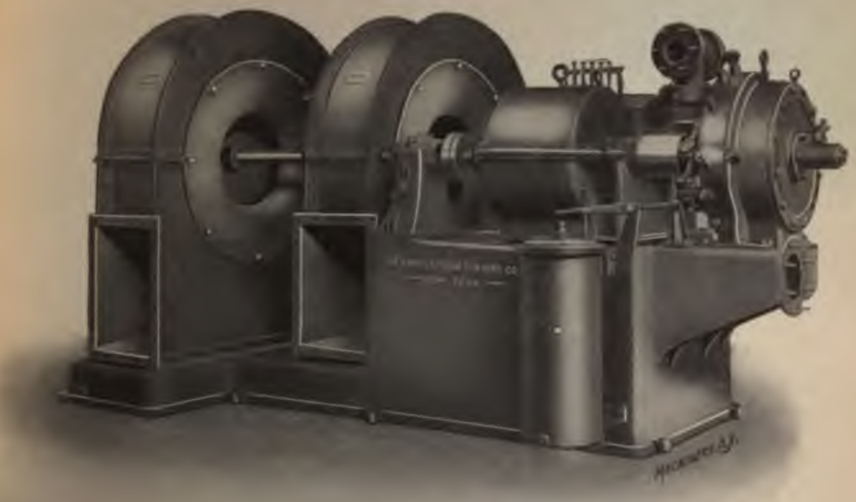


Fig. 11. Application to Blowers of the Sirocco Type.

Blower Sets.—Another field where the turbine is well adapted is for direct connection to blowers for water pressures between 4 and 21 inches. The high velocity of the turbine makes it feasible to use such blower units for locations and pressures where the positive acting impeller blowers have been employed. The turbine is used in connection with blowers of the Sturtevant and Sirocco type. When directly connected to the blowers, the whole forms a compact unit, and eliminates the trouble from tight belts and heated bearings met with in attempting to use blowers for high pressures.

CHAPTER IV

THE PELTON AND SIMILAR TYPES.

Simple impulse wheels of the Pelton type have been experimented with extensively by Professor Rateau of Paris and Professors Riedler and Stumpf of Berlin; although Rateau has now abandoned this type in favor of his multicellular turbine, and the

Fig. 1. Wheel of Rateau's Turbine.

manufacturers of the Riedler-Stumpf turbines appear to give preference to compound turbines of later design. The fact, however, that the Pelton water wheel, commonly known as the "Hurdy-Gurdy" wheel, and its later rival the Doble turbine, have been so successful in American water-power plants where there is a high head and a high velocity of water jet at the wheel, makes it seem probable that steam turbines similar in principle to the Pelton water wheel will be experimented with in this country to a considerable extent.

Rateau's Simple Impulse Wheel.—Fig. 1 is an engraving of Rateau's wheel, a line drawing of which was shown in the patent review of the second chapter. In Fig. 2 are sectional drawings, showing the details of the turbine itself as constructed from Rateau's designs. A number of turbines similar to this were built, the one represented being direct-connected to a blower.

*The Riedler-Stumpf Wheels** were invented by Professor

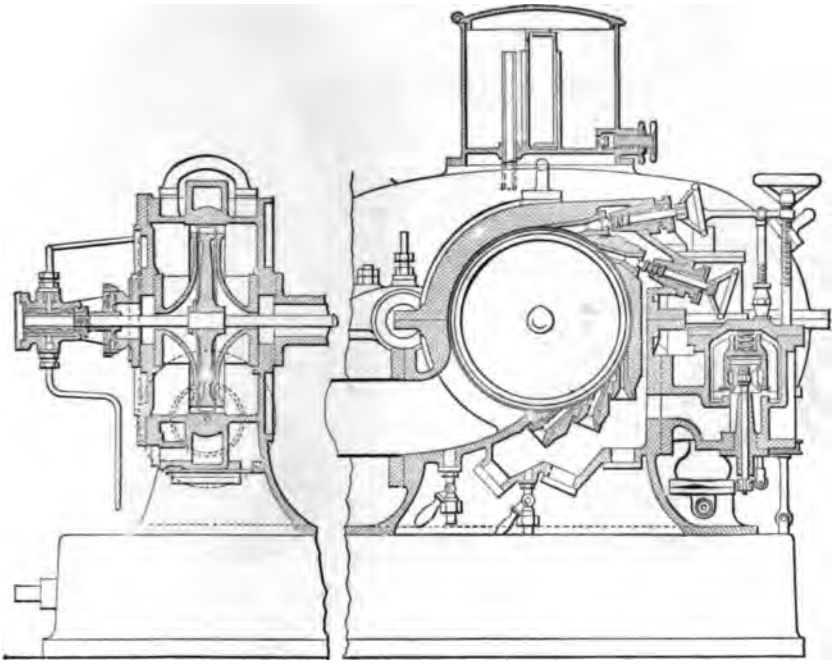


Fig. 2. Rateau's Simple Impulse Turbine.

Stumpf and developed with the assistance of Professor Riedler. A wheel for a 2,000-horse-power turbine is shown in Fig. 3 and the nozzle ring for the same in Fig. 4. In a turbine of so large a size it was necessary to use a complete ring of nozzles, but in machines of smaller power one or more segments containing nozzles are all that are required. The groups of nozzles are connected with a central distributing chamber by means of radial tubes so arranged

*Described in a paper read by Professor Riedler in Germany. See "Machinery" for February, 1904.

that steam may be admitted to one or more of the groups as desired. The buckets are cut into the rim of the wheel, but the nozzles are placed obliquely upon one another in a ring which surrounds the wheel. The nozzles are of the De Laval type made of nickel steel, but are square in cross-section. They are produced first in the form of round tubes and the diverging parts are then

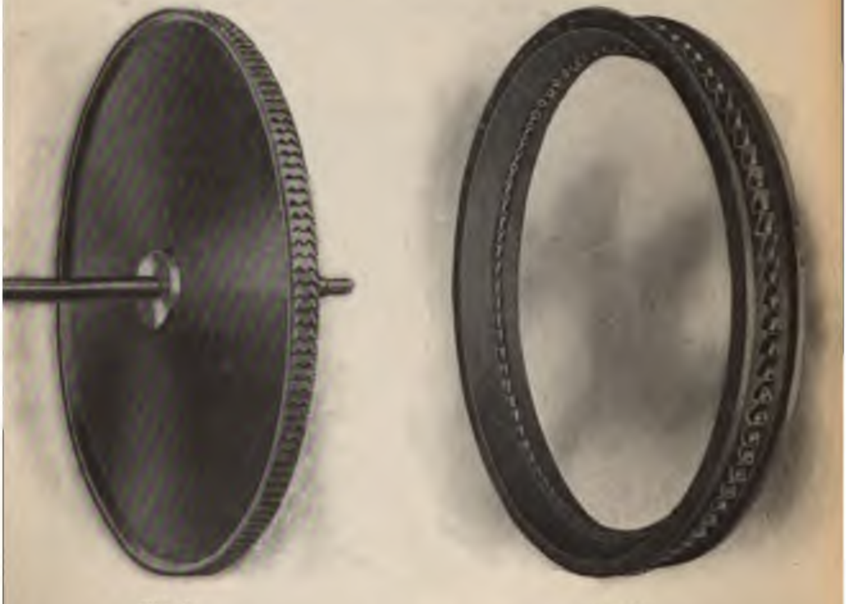


Fig. 3. Fig. 4.
Wheel and Nozzle Ring for Riedler-Stumpf Turbine.

drawn out square and finally cut off obliquely. The details of the wheel are indicated in Fig. 5, which shows sections of the buckets. It will be noted that the buckets are so formed as to overlap each other something like the shingles of a roof, instead of being placed one in front of another as in a Pelton water wheel. They are designed to reverse the steam jet through the whole angle of 180 degrees. In order to reduce the velocity of rotation below that obtained in the De Laval wheel, Professor Stumpf increased the diameter of his wheels from six to nine feet, and he also found it expedient to abolish the flexible shaft by giving unusual atten-

tion to the balancing of the wheels, which are in the form of flat disks. Professor Riedler states that these disks may be balanced so that the center of gravity will come within .004 of the geometrical center; also, that wheels $6\frac{1}{2}$ feet in diameter will have a factor of safety of 5 when running at 3,000 revolutions per minute. As a result of careful tests he is satisfied that nickel steel disks of this size may be procured which are practically free from internal strains.

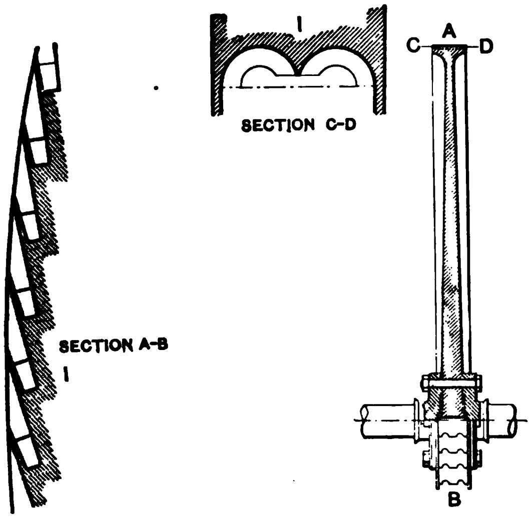


Fig. 5. Sectional Views of Riedler-Stumpf Wheel.

A Stumpf Patent, issued in 1903, specifies double U-buckets with a dividing ridge milled out from the solid rim of the wheel as described above. Fig. 6 is from the patent drawing. He proposes to use a series of guide passages having the general contour of the dotted line shown at *A,* which gather up the steam after it has issued from the two sides of the wheel and return it in a single, solid stream at the center, where it impinges a second time against the blades of the same wheel. In Fig. 7 is shown a modified form of the guides having partitions extending in the direction in which the steam flows, for the purpose of insuring an equal division of the steam jet over the whole breadth of the return buckets, with a view to preventing choking of the steam at the point where it strikes the wheel vanes.

It is pointed out by Professor Stumpf that steam, in reversing its direction of flow in a turbine wheel, acquires sufficient centrifugal force to increase its pressure, frequently by a considerable

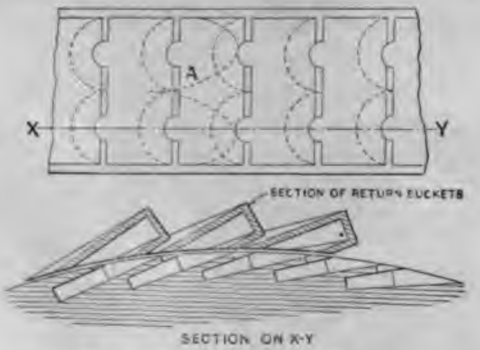


Fig. 6. Subject of Stumpf Patent.

amount, so that in leaving the vanes there is a sudden explosive expansion of the steam, causing a scattering of the jets. By catching the steam in the return buckets as it leaves the wheel, and bringing the streams together in a solid jet again at the center, he aims to overcome this action and to produce a more efficient type of compound turbine. His claims are broad ones, applying to the combination of admission nozzles, a turbine wheel with double buckets and double return buckets.

In Figs. 8 and 9 are two arrangements that have been adopted for applying the features of the Stumpf patent. The wheel, instead

Fig. 7.

of having double U-buckets, has a single U-shaped bucket. The steam flows from the nozzle and strikes against one side of the buckets, then passes around to the other side and escapes to the guides, which again change its direction of flow and cause it to

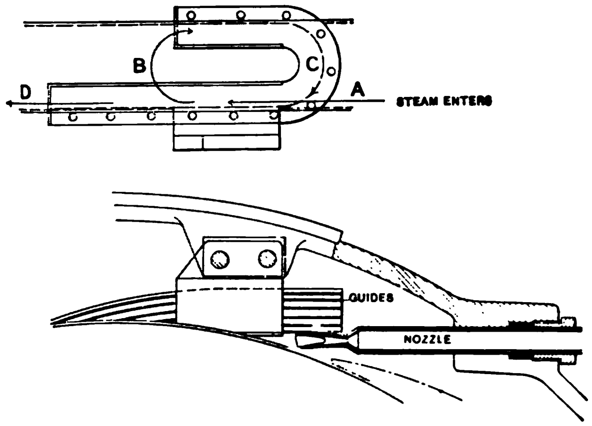


Fig. 8. Showing Arrangement of Guides.

impinge a second time against the wheel. The course of the steam is indicated by the letters *A B C D* in the illustration.

In Fig. 9 is the arrangement where there are double U-buckets. The steam enters through the nozzle in the direction of the arrow *A*, the jet divides when it strikes the buckets, flowing in the direc-

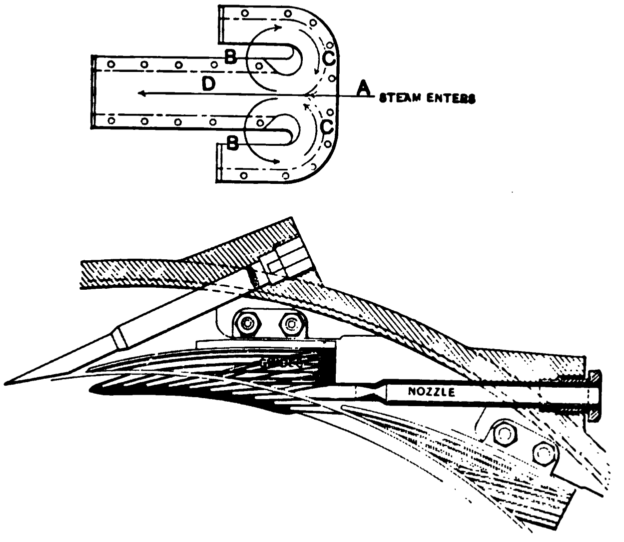


Fig. 9. Modified Arrangement of Guides.

tion indicated by the arrows *B B* and *C C*, and finally impinges the second time against the buckets in the direction of the arrow *D*.

In a subsequent chapter further reference will be made to the Riedler-Stumpf turbine, in which the compound principle is applied by using two or more wheels instead of a single wheel as here described.

Claims Made for the Pelton Type.—The best presentation of the claims for a Pelton type of wheel for steam turbine purposes has been given by John Richards in a paper before the Technical Society of the Pacific Coast in May, 1904.* He argues that steam impulse wheels as usually built are at fault because the blades are curved in one plane only, and consequently have but one correct position in the jet throughout the whole arc of their movement; and furthermore in nearly all cases are cut out of solid metal, and have angular or imperfect corners. Whether the principles enunciated will hold when using an elastic fluid like steam instead of an inelastic fluid like water can only be told by experiment. The tendency of the steam jet is to break up into spray and eddy currents, whereas a water jet will "hang together" for a longer period. This act may have an important bearing on the question of the spacing of the buckets in a Pelton type of wheel when used with steam.

The following is extracted from Mr. Richards' paper, beginning with his objections to the type of bucket employed in impulse steam turbines as now constructed, such as in the De Laval:

First. It increases the weight and number of the buckets about fivefold in the attempt to secure impingement of the steam jets normal to the straight faces of the buckets.

Second. It distorts the course of reaction from a possible angle of 15 degrees to an angle of 20 to 30 degrees required to secure clearance.

Third. It makes necessary a side application of the jet, introducing lateral stress on the wheels and inducing vibration.

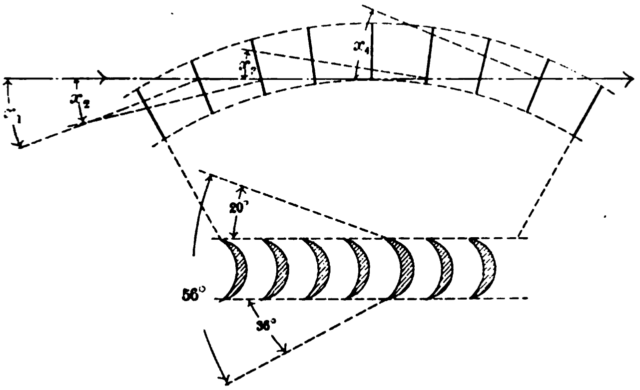
Fourth. It augments, in proportion to the added number of buckets, the amount of fluid friction. Not to include the resistance of corners.

The number of buckets is an important matter. It is a sequence of the angle of impingement, and this again is a sequence of the bucket's shape, as will be shown further on. The surface or fluid friction, which offers a con-

*Published in the Journal of the Association of Engineering Societies, Philadelphia, September, 1904.

siderable resistance and loss, is in proportion to the number of buckets employed, and should be considered in this connection.

Most of the steam turbine buckets now made have angular corners, and, when there are not such corners, the end walls of the buckets are so distant from the jet as to lose reactive effect in that direction. We long ago learned to keep water out of sharp corners in hydraulic practice.



Figs. 10 and 11.

Figure 10 shows how the line of impingement varies in respect to the straight faces of radial buckets, and there is no way of securing impingement even approximately normal to the straight faces, except by employing a large number of buckets set close together. The result is much the same whether the jets be applied tangentially or on the side, as shown in Fig. 11, where the angle of entrance is 20 degrees and that of discharge 36 degrees in conformity with the practice of the De Laval company.

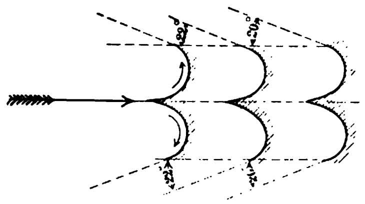


Fig. 12.

The trend of practice in tangential water wheels has been to wider spaces between the buckets, better angles for discharge, and, recently, to uniformly curved buckets, as hereinafter explained.

In Fig. 11 the entrance and discharge angles embrace an arc of 56 degrees, which, by reducing the number of buckets, could be reduced to 36 degrees or

less if the problem of oblique impingement were out of the way. Fig. 12 shows spacing for tangential buckets to secure an easy discharge at 20 degrees.

In the Riedler-Stumpf turbines, the angle of discharge is 180 degrees. In other words, the discharge is opposite the jet, but this calls for increased surface, more width and weight for the revolving member, and expensive work in construction, which are hardly offset by countervailing advantages, and which certainly prevent a cheap and general manufacture of the machine.

Mr. Richards then contends that buckets of steam turbines should be curved in all planes approximately as shown in Fig. 13, taken from a form of water buckets of an advanced type by W. A.

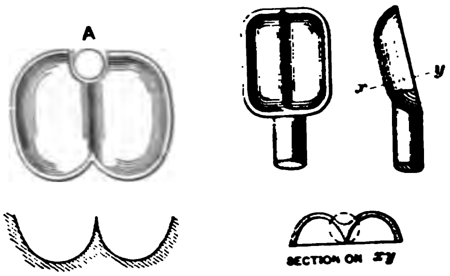


Fig. 13. Buckets Suggested by Richards.

Doble of San Francisco. These are of double concave or cup form, in order to permit direct and balanced impingement at the various angles in which they are presented to the jet, and have a central dividing wedge to permit tangential application. The bucket is notched at *A*, following the construction of certain water wheels, permit the passage of the jet beyond and through the buckets they come into position, so that it will impinge against the kets in advance which have reached a position where the jet act upon them efficiently. He estimates that about one bucket each 8 degrees of arc will be sufficient for wheels from 20 to 40 ches diameter. This is less than one-fifth the number now employed for wheels having the ordinary type of buckets.

Zoelly's Patents.—In 1900 a patent was taken out by Heinrich Zoelly for a turbine wheel of the Pelton type, but with radial arms, the outer ends of which serve as vanes for the wheel. These arms decrease in cross-section as they approach the periphery of the

wheel and thus are proportioned to resist the stresses due to centrifugal force, which are greater near the center. The first claim of this patent is for "the combination in a turbine wheel of radial buckets separated from each other for a part of their length, each bucket having its receiving face channeled for the greater portion of its length, and a pair of flat disks inclosing said buckets from their inner ends for a greater portion of the length of the channeled part of the buckets." In Fig. 14 *A* and *B* are two sectional views

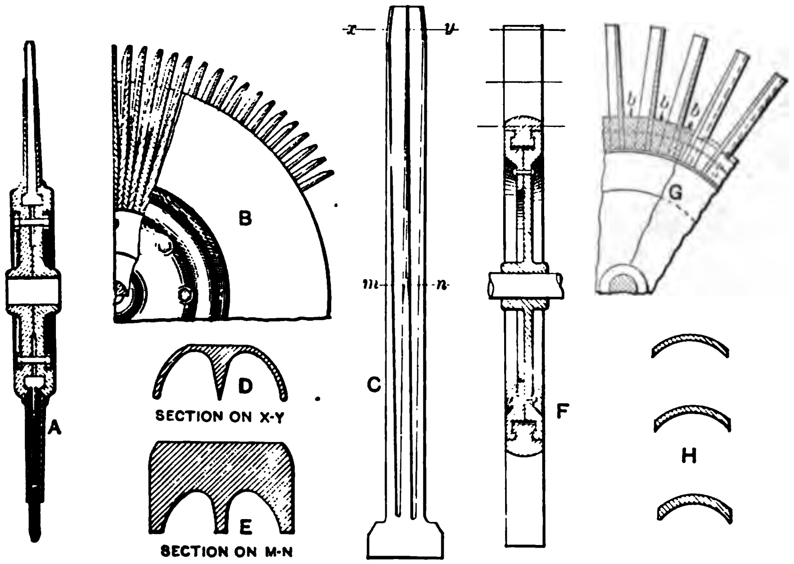


Fig. 14. Wheels Patented by Zoelly.

of the wheel, *C*, an enlarged front view of one of the blades, and *D* and *E*, enlarged sections of the blade on the lines *x y* and *m n* respectively.

At *F* and *G* are details of a wheel now used in the Zoelly turbine described in a subsequent chapter. In this type, patented in 1903, the steam is directed against the blades on one side of the wheel and escapes on the other side. Sections of the blades are shown at *H*. The first claim is for a "turbine blade constructed with a gradually increasing longitudinal thickness and a longitudinal cavity of substantially uniform depth." Some of the other

claims relate also to the method of clamping the blades in position, and the use of spacing blocks, *b b,* between them.

Richards' Patent.—The idea presented in the Zoelly patent of 1900 is carried a step further in a patent issued to J. Richards for a wheel in accordance with his ideas. The wheel consists of buckets, *B,* light in weight and drop forged on the ends of radial arms, which are attached to a central nave by pins inserted between the arms, as indicated in the illustration. The buckets are spaced further apart than is usual in turbines, because they are designed to be concave and reactive through a considerable angle of rotation, and thus will absorb the energy of the jet sufficiently throughout

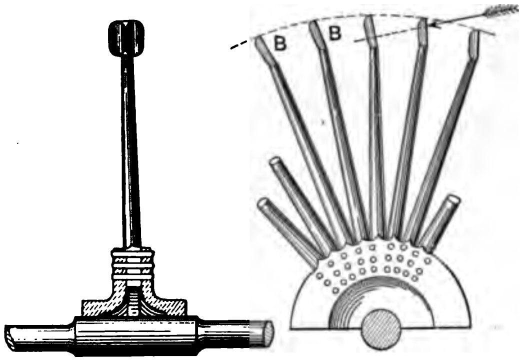


Fig. 15. Turbine Wheel Proposed by Richards.

this range. The arms of the wheels are not covered by plates at the sides, as in the Zoelly design; the intention of the inventor being that the inside of the turbine casing shall be machined smooth, and the steam allowed to rotate with the wheel within the casing. The claim is in substance for a wheel having a single hub, of a diameter within the zone of disruptive centrifugal strain, with equidistant radial sockets formed therein; strong radial stems fastened in the sockets; and concave reactive buckets integrally formed on the extremities of the stems.

Turbine Designed by John Richards.—In Fig. 16 is a turbine proposed by John Richards and patented by him in 1903. In this turbine he uses a wheel like that outlined above. The casing for the wheel is finished smooth inside and the steam is supposed to rotate with the wheel in the casing. The gearing of transmission

is inclosed with the motor wheel and operates within the vapor contained in the casing. The three reasons for this are: (1) That he believes the wheels will wear better when steam lubricated; (2) that noises, if present, will be abolished; (3) that by this construction packing glands on the spindle of the motor wheel may be abolished. The journals for the wheel spindles are to be hardened and ground, mounted in pivoted split shells of cast iron, and like the gearing, exposed to the vapor of the wheel chamber, which is at low pressure.

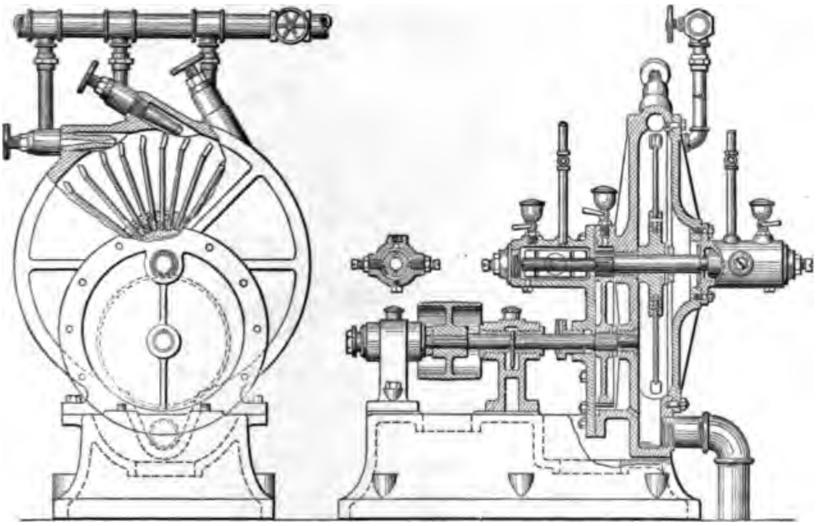


Fig. 16. Turbine Patented by John Richards.

Mr. Richards is an advocate of simple impulse wheels with gear transmission, because such types are more cheaply constructed than elaborate compound turbines. In his paper before the Technical Society of the Pacific Coast, already referred to, he gives numerous examples within his experience, of high speed gearing and rapid running machinery, which have given service during long periods of time.

Levin's Turbine.—In 1904 a turbine was patented by A. M. Levin in which the steam is expanded completely in a nozzle before impinging against the buckets of the wheel and then is used several times in succession upon the blades of the same wheel.

team is expanded in the nozzle, N, which projects it against one ide of the semi-circular buckets, B, of the wheel. The steam basses around these buckets and is projected outward against the curved surface, C, of the casing twice in succession, which at each time redirects the steam against the buckets of the wheel. The curved surface of the casing is stepped so that the portion at C^1 may be brought nearer to the wheel, and when the steam reaches the buckets at B^1 it is projected against the surface at C^1. As drawn, the arrangement is designed for a wheel having a peripheral speed of one-tenth the initial velocity of the steam since the steam is projected against the buckets five times in succession. The buckets are semi-circular in form.

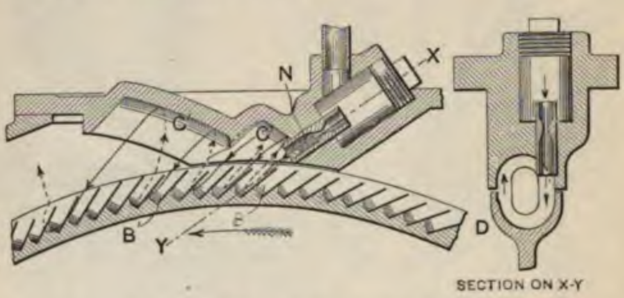


Fig. 17. Experimental Turbine of Levin.

The steam proceeds in a succession of helical whirls after leaving the nozzle, and it is necessary that the steam should be completely expanded in the nozzle so that it will be at constant pressure, but have a decreasing velocity after leaving the nozzle. The first claim for this wheel is for a "multiple impulse turbine, comprising a wheel having a row of buckets, an expansion nozzle delivering into said buckets, and a stationary reversing guide extending from said nozzle over a number of said buckets, to form a space open end to end within which the motive fluid proceeds in a helical whirl and is successively projected against the buckets of said wheel."

Mr. Levin built an experimental turbine on this order which was described in "Power" in May, 1904. One of the interesting features is a widening of the semi-circular grooves constituting the wheel buckets and the guide surfaces at the points where the steam

leaves the grooves, as clearly shown in Fig. 17 at *D*. This is because the steam becomes compressed in passing around the curved surfaces, and at the points of escape, the passages are widened to allow the steam to reëxpand to its previous volume. In reference to frictional losses, Mr. Levin states that his tests indicate they are not of prohibitive importance, nor has he found indications of wear even when moist steam has been used. On the contrary, after having run the wheel with moist steam, the buckets were always coated with a fine bluish film, apparently derived from oil carried over from the boiler.

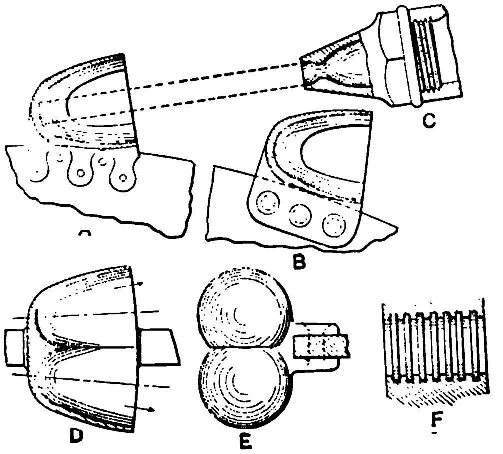


Fig. 18. Buckets of Kerr Turbine.

C. V. Kerr's Turbine.—Another patent is that issued to Mr. C. V. Kerr for a compound impulse wheel of the Pelton type. The buckets are made of drop forgings and of such a shape that they may be bored out perfectly smooth by means of a special reamer. Accordingly, each recess in the bucket on either side of the dividing wedge has a contour representing a surface of revolution. Sketches of the buckets appear in Fig. 18. The curves of the interior of the bucket in a transverse direction must obviously be circles in whole or in part, as shown at *E*, whereas the longitudinal section will show curves elliptical in shape.

The buckets are attached to steel disks, and in order to withstand the great strain he prefers to attach them, as shown at *A*, by dove-

tailing and upsetting the interlocking parts, or else by electric welding. Another construction suggested is by riveting, as shown at *E*. Expansion nozzles are used, of the form indicated at *C*, Fig. 18, the tip of the nozzle being rather short inasmuch as the turbine is divided into stages and only a portion of the pressure of the steam has to be reduced at each stage. Each nozzle of the turbine is controlled by a hand valve.

The Kerr turbine is being developed by the Kerr Turbine Company, Wellsville, N. Y. In Fig. 19 is a diagram illustrating the arrangement. Steam flows through a series of nozzles and

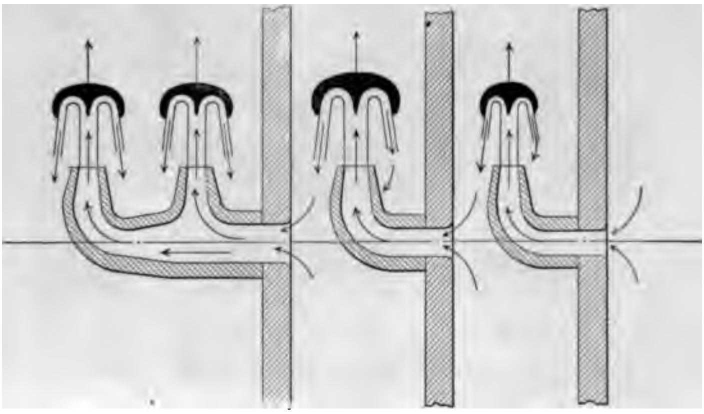


Fig. 19. Showing Principle of Kerr Turbine.

impinges against the cups of the first wheel, which is located in a compartment by itself. It then flows through another series of nozzles and impinges against the cups of a second wheel in a second compartment, the cups being enough larger to accommodate the increased volume of the steam at the lower pressure. It again discharges into a third chamber, in this case against two wheels. This arrangement is followed throughout, there being an increase either in the size of the cups or in the number of wheels as the low pressure end is reached. This design provides for manufacture in standard parts, because by combining the units turbines of widely varying powers can be constructed without increasing the size of the individual parts. The governor is of the throttling type.

CHAPTER V

COMPOUND IMPULSE TURBINES—MULTICELLULAR TYPE.

The Rateau Steam Turbine.

History.—The practical work of M. Rateau on steam turbines began in 1894 when he constructed the first simple impulse turbine, which has already been described. He soon abandoned the single-wheel turbine, however, in favor of the system with multi-

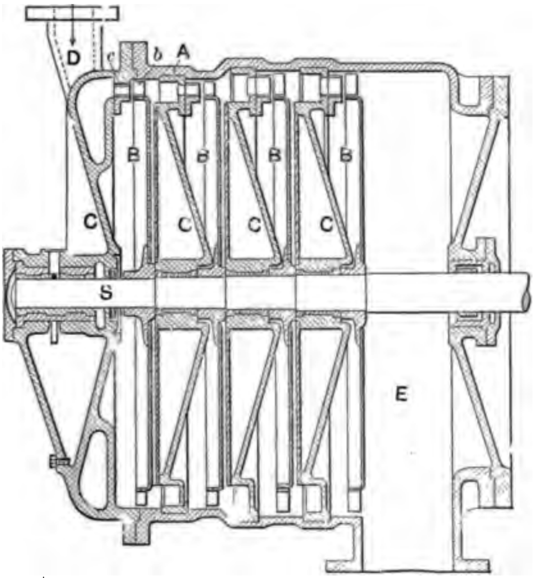


Fig. 1. Diagram Showing Principle of Rateau Turbine.

ple wheels constructed on the impulse principle. His ideas shaped themselves about 1897 and the following year the firm of Sautter, Harlé & Co., Paris, began the construction of a 900-horse-power machine which was experimented with until about 1901, when the machine was brought to a high degree of perfection, and since which time many others have been built. A modified type of the Rateau turbine is also built under patents of the inventor at

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Fig. 17.

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whirl and is s
wheel."

Mr. Levin

The principle of this turbine as thus far outlined is in no way different from that of (Chapter II.), but Rateau has introduced an arrangement of the vanes which is new and is covered by this patent. In Fig. 8 the first set of guide blades is shown at A. These are five in number and are arranged in, say, groups about the periphery of the first disk. After passing

next set of guide blades is reached at *B*, er number to accommodate the increasing d these blades are arranged so as to extend rection in which the steam flows, as shown. es the first wheel it will be carried along a rotation of the wheel before discharging er, and a portion of the next set of guide ed in advance of the previous set. At *C* the l more, and finally a point will be reached ll extend around the full periphery of the ge of this arrangement over one in which ed to the turbine around the whole periphery

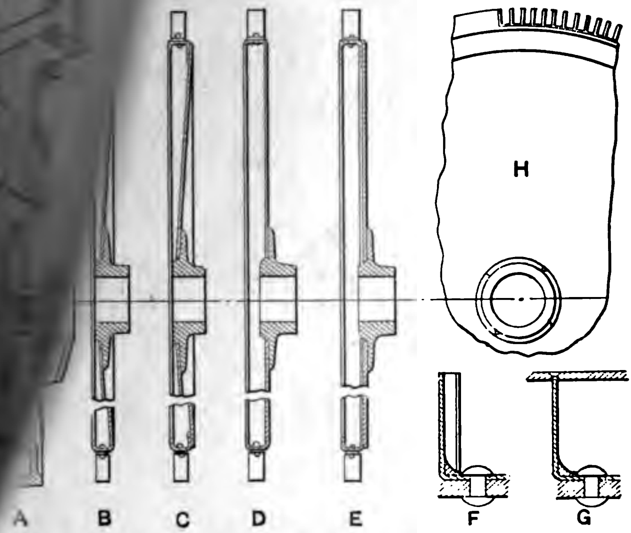


Fig. 3. Construction of Rateau Diaphragms and Wheels.

as the volume of the steam at the admission point is small nes would necessarily have but little radial depth at that if they comprised a full circle, and there would be excessive on of the steam when flowing through them. In the Rateau ngement the steam passages are deeper and the volume of am passing is large, in proportion to the rubbing surfaces of e vanes. The reference to this in the claims of the patent is a llows: "* * * distributors arranged in the membranes t

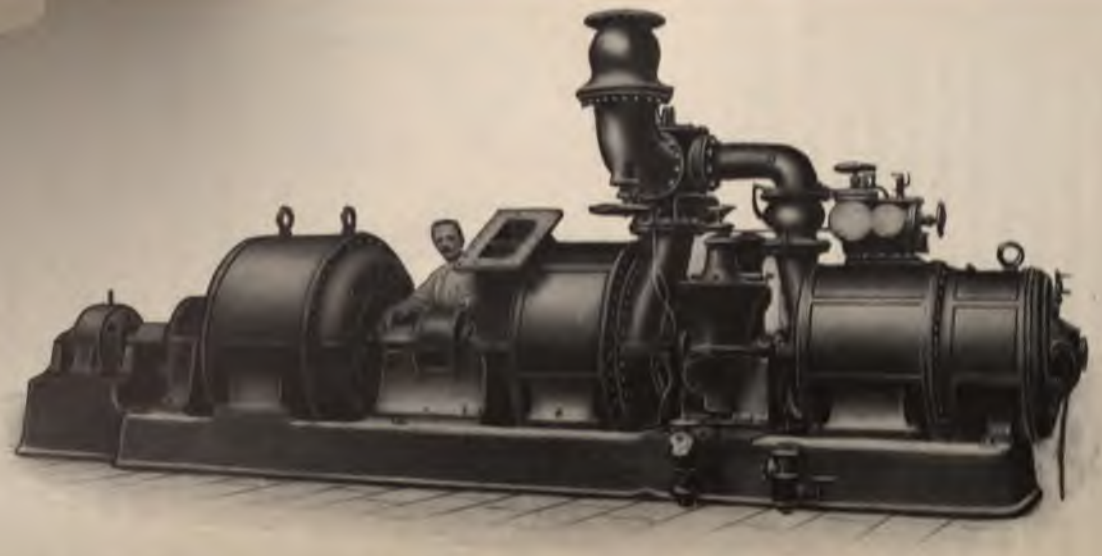


Fig. 4. Rateau Turbine Driving Alternator.

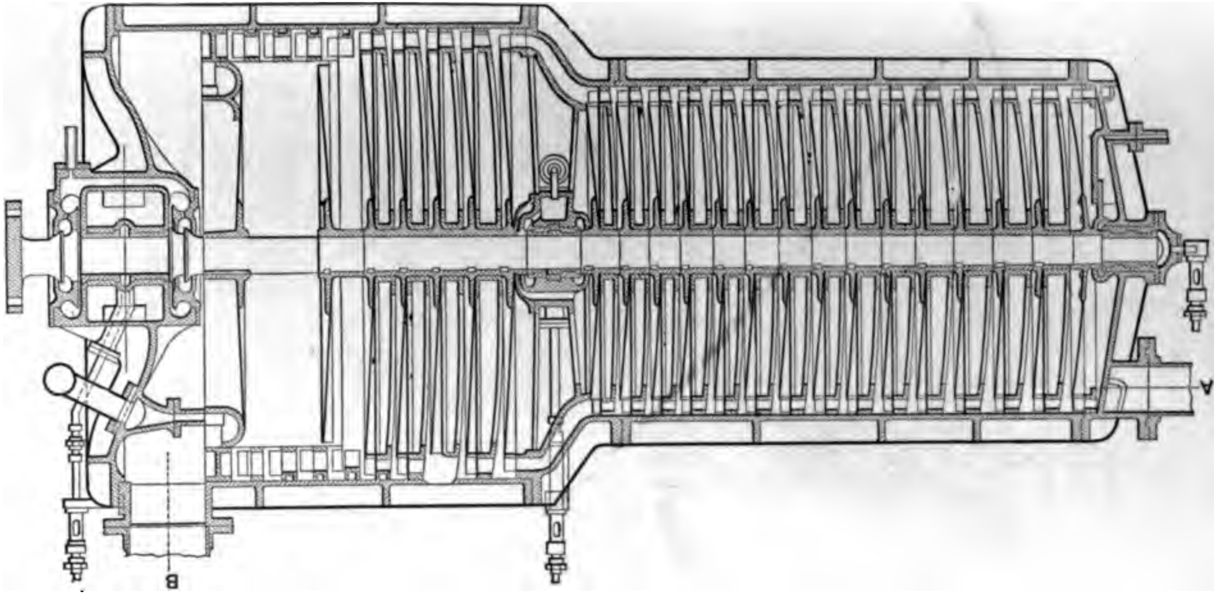


Fig. 5. Sectional Elevation of Rateau Turbine. Steam Admission at A, Exhaust at B.

direct the motive fluid directly upon the paddle blades, and said distributors increasing in width, and overlapping each other successively at one end and not at the other." Rateau also introduces features of construction on which claims are made, but which are in no way tied with the blade arrangement mentioned above. Some of these are shown in Fig. 3. At *A* is one of the diaphragms

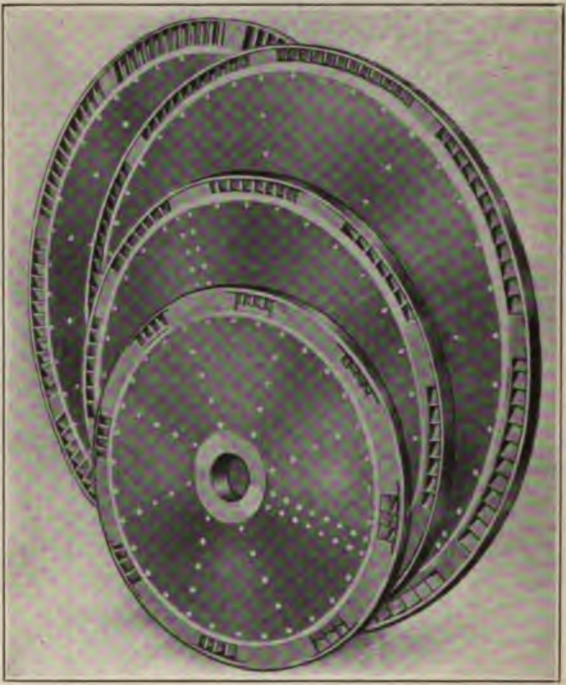


Fig. 6. Group of Diaphragms or Distributors for Turbines of Different Sizes.

containing the guide vanes. At *B, C, D,* and *E* are typical wheels consisting of steel disks either flanged around their peripheries or else with annular channels riveted to their peripheries. In the first two instances, *B* and *C,* the disks are dished to add to their lateral strength and in the last two they are flat.

The vanes, which are curved suitably at the points where the steam strikes, are bent on an angle and riveted to the circumference of the disk. At *F* and *G* are enlarged details of the vanes, the

second one showing a band or shroud riveted to the outer circumference.

Practical Notes.—In designing the turbine, Professor Rateau attempted to attain the three following main objects: 1. A high mechanical efficiency together with as low an angular velocity as

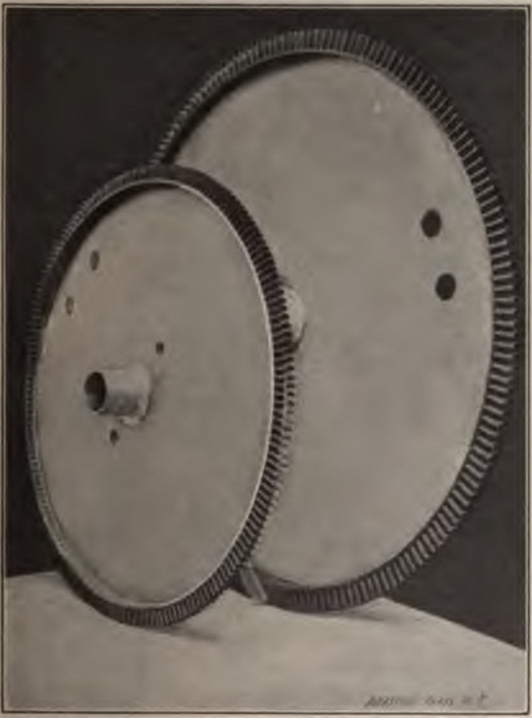


Fig. 7. Pair of Wheel Disks.

sible. 2. A large and at the same time non-injurious clearance ween the fixed and moving parts. 3. The least possible weight the whole machine, and especially of the rotating parts. In . 6 is shown a group of diaphragms or distributors for tures of different sizes. The guide vanes are arranged in groups ut the periphery and the number of openings about each group reases about the exhaust end of the turbine until they finally end around the whole periphery. As in an impulse turbine the

expansion of the steam is complete in this distributor, so that the steam acts upon the wheel in virtue of its velocity, and as the wheel vanes are symmetrical in shape, end thrusts are practically eliminated. The shaft passes through the diaphragms in bushings of anti-friction metal. A pair of wheels is shown in Fig. 7. These are constructed as indicated in sketch Fig. 3.

The bearings of these turbines are external and by means of a system of spring packing are kept perfectly tight. The speed is controlled by a centrifugal governor acting by varying the pressure of the steam delivered to the turbine. By means of a by-pass in the main steam pipe it is possible to deliver steam of full pressure both to the entrance of the turbine and to a point in the machine nearer to the condenser, thus enabling a higher power than the normal amount to be produced by the machine, much in the same manner that a compound engine may be used with full pressure steam and low pressure cylinders.

The Zoelly Turbine.

A steam turbine now attaining a prominent place abroad is the Zoelly turbine which has been developed and is now manufactured by Escher, Wyss & Co., Zurich, Switzerland, the famous manufacturers of hydraulic turbines, water-wheel governors, etc. A number of large German firms, of which the Krupps are one, are reported also to have formed a syndicate for the manufacture and sale of this turbine on a large scale.

The general arrangement of the Zoelly turbine is evident from the half-tone illustration, Fig. 8, and the line drawing, Fig. 9, most without explanation. The turbine is divided into two parts, cased separately, and placed far enough apart to permit a bear-g to be located between them for supporting the shaft at the nter. In Fig. 8 the top of the casing of the low-pressure com-artment is lifted, exposing the wheel blades to view, and in Fig. 9 the same compartment is shown in section. The construction of the high-pressure section is entirely similar to the low-pressure, except that the steam passages have less area.

There are ten rotating wheels constructed in the form of circular disks, attached to the same shaft and carrying curved blades on their peripheries. For each wheel there is a set of guide vanes

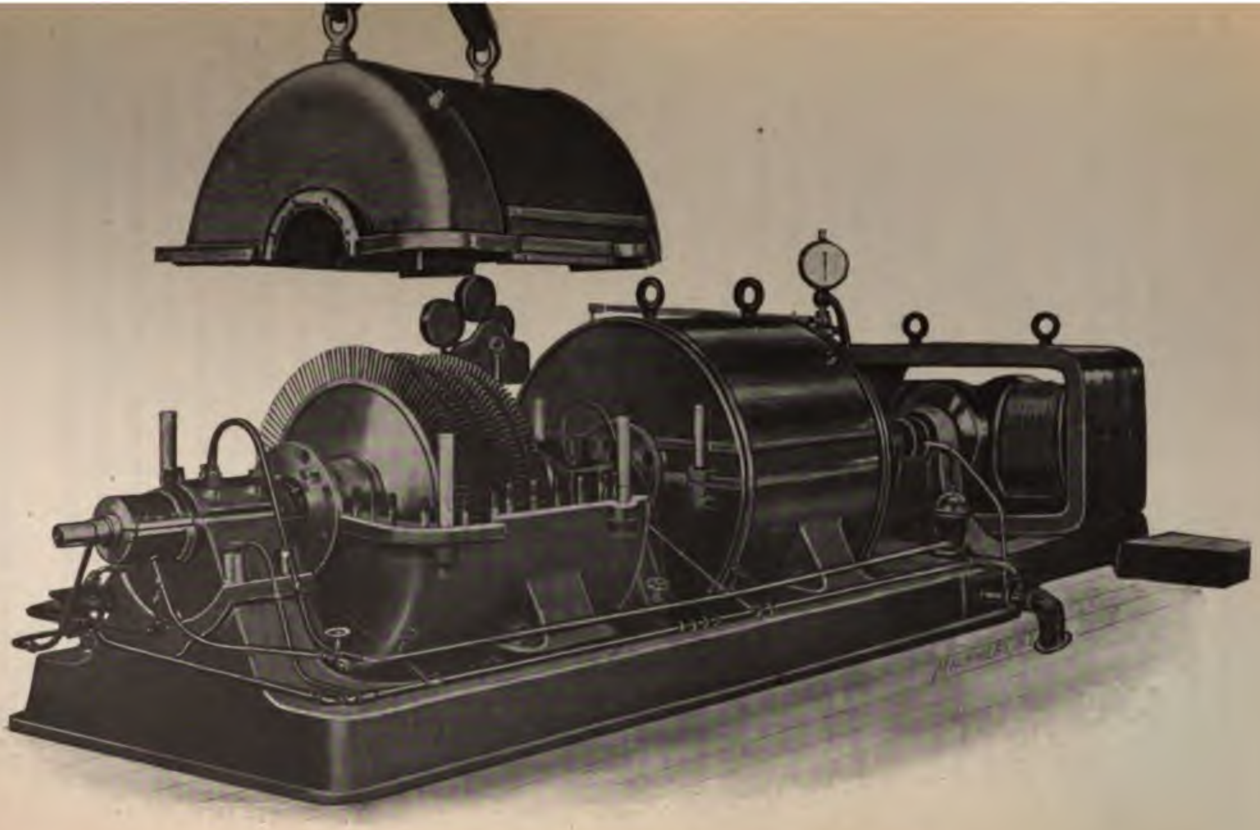


Fig. 8. Zoelly Turbine with Top of Low-Pressure Casing Removed.

for directing the steam against the rotating blades. These guide vanes give the steam the proper direction of flow and allow it to expand a certain amount as it passes through the guide passages, their function being the same as that of the steam nozzle in the De Laval turbine and the guide passages of the Rateau turbine. Each wheel rotates in a chamber by itself, the walls of which are formed by the disks to which the guide vanes are attached. The steam enters at *A*, Fig. 9, and passes through a throttle valve operated by the governor, to the high-pressure compartment. Here it flows through the first set of guide passages, and impinges against the blades of the first wheel. The guide passages permit the steam to expand to the somewhat lower pressure of the first chamber and thus partially convert its potential energy to kinetic energy, which is mainly given up to the rotating wheel, since the steam leaves the wheel at a low velocity. The steam now passes through the passages of the second guide disk, expands to a lower pressure and is directed against the blades of the second rotating wheel in the second compartment, where it again gives up its kinetic energy. When the last step is reached in the low-pressure compartment the steam finally exhausts, either into a condenser or into the air.

Guides and Wheel Vanes.—At the beginning of the high-pressure section, the guide vanes occupy only a part of the periphery of the turbine, but toward the end of the low-pressure part they extend around the whole circumference. It will be noted in Fig. 10 that the passages through the guide vanes have parallel sides; that is, the walls do not diverge as in the nozzles of the De Laval turbine. This construction is based on the well-known fact that steam will expand and convert its available heat energy into kinetic energy, or the energy of motion, by flowing through a nozzle having straight, parallel sides, provided the final pressure is not less than .58 of the initial pressure; whereas, if the final pressure is less than this, the walls of the nozzle must diverge in order to fully expand the steam. In the Zoelly turbine the expansion occurs in successive steps and the pressure does not drop sufficiently at any one step to make guide passages with diverging sides necessary.

It will be noted further, from Figs. 9 and 10, that the passages

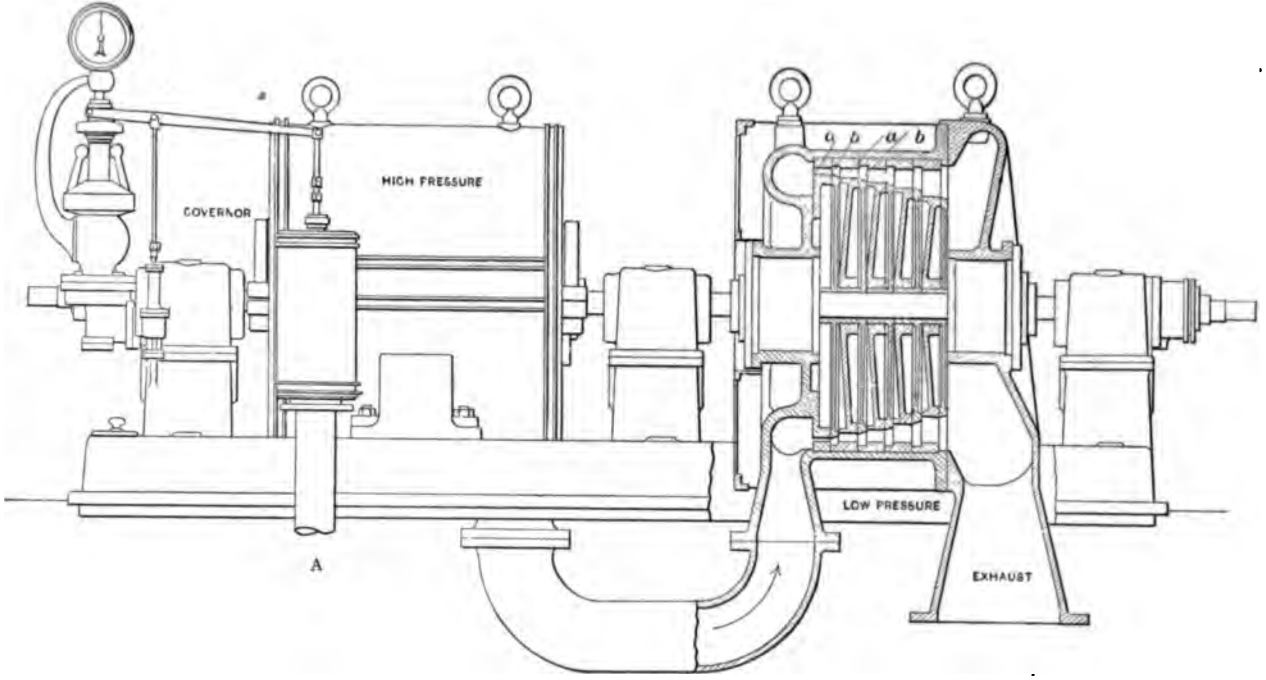


Fig. 9. Elevation of Zoelly Turbine.

through the wheel blades have their inner sides inclined, producing channels of gradually increasing area. This is not, however, to allow for expansion. In this turbine, as in others of the impulse type, the pressure of the steam does not change in passing through the rotating wheel. The pressure is uniform throughout the chamber in which the wheel turns, making a drop in pressure in passing through the wheel impossible, and hence the only effect of

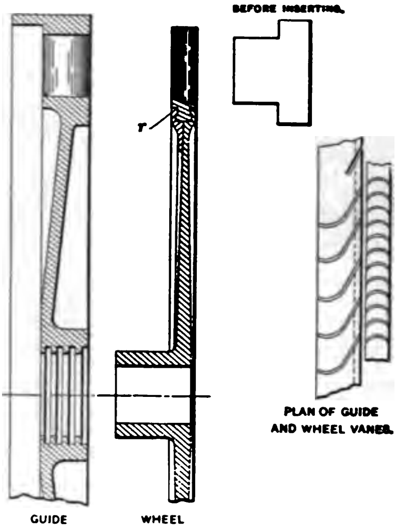


Fig. 10. Section of Diaphragm and Wheel.

the sloping sides of the wheel passages is to cause the steam to flow smoothly, without eddy currents, into the next guide passages, which are of larger area than the ones preceding. The conditions under which the steam flows through the wheel are entirely different from those which influence the flow through the guide vanes, for the latter have a higher pressure on one side than on the other.

Enlarged sections of the wheels and guides are shown in Fig. 10. The principal difference between the Zoelly turbine and others of similar type, such as the Rateau, lies in the construction of the details of the turbine wheels. These are designed to permit

a high rotative speed without straining the material, which makes it possible to reduce the number of steps used in the turbine to a minimum. The wheels are forged of one piece, including the hub, from Siemens-Martin steel and a T-slot is machined out of the circumference of the disk to hold the blades and their distance pieces. One side of the slot is closed by a wrought iron ring, *r*, which is riveted to the disk. The blades are radial, and made of nickel steel, highly polished, which resists erosive action very effectually. The cross-section of the blades decreases as the radius increases, thus reducing the stresses due to centrifugal force to a minimum. The blades and distance pieces are milled by special machinery.

Wheels and Disks.—The disks which hold the guide vanes and separate the turbine into compartments, are bored out to receive the hubs of the turbine wheels. As there is a higher pressure on one side of each disk than on the other, it must be steam tight to prevent leakage and strong enough to prevent deflection. The only place where leakage can occur is at the center, where the bore of the hub must be loose enough for a running fit over the hub of the wheel. Annular grooves are turned in the bore to reduce this leakage, on the same plan that grooves are sometimes turned in the surface of a pump plunger for the same purpose. The guide disks are divided on a diametral line, with their top halves bolted to the top of the casing and their lower halves to the base so that the top casing and guides can be lifted off, exposing the wheels. The joints of these disks are ground to a close fit, to prevent leakage.

The casings for the high- and low-pressure parts of the turbine are mounted on the bed plate by brackets placed midway between their ends to prevent distortion from expansion, due to the heat of the steam. There are also large clearance spaces between the guide vanes and wheel blades, made possible by the fact that there is but small tendency to leak at these points, as explained above. This reduces the likelihood of the blades being displaced sufficiently, through the heat of the steam, to cause rubbing when the turbine is running.

Governor.—In Fig. 11 is a detail drawing of the governor, which controls the turbine by throttling the steam. A centrifugal

governor acts on a relay valve, *m*, and connects one side or the other to pipes *a* and *b*; *a* being a pipe leading from a reservoir full of a liquid, such as oil or water, under pressure produced by a rotary pump; and *b* a return leading to the suction well of the pump. The two pipes *e* and *f* connect each end of the valve to the cylinder *g*, which is located on top of the main throttle valve.

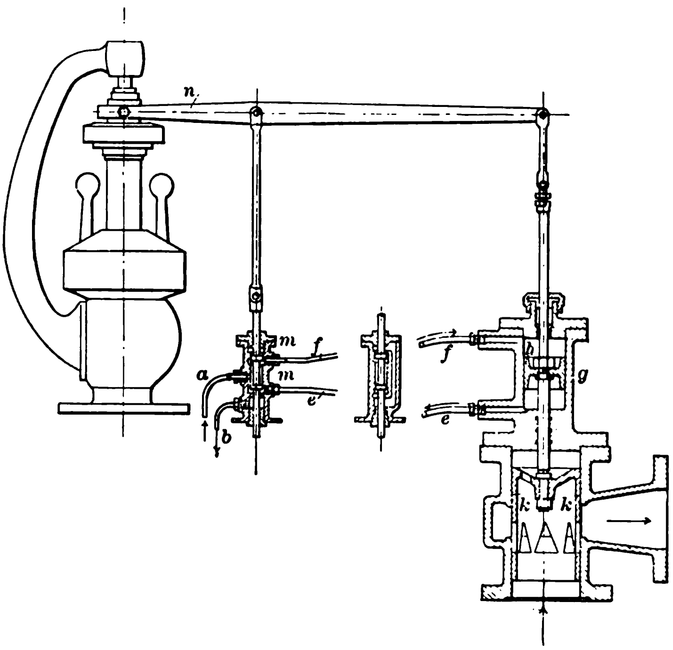


Fig. 11. Governor of Zoelly Turbine.

The moving part of this valve is attached to the same valve stem as the piston *h*, in the cylinder *g*. If the load on the turbine is decreased, the resulting increase in speed raises the governor lever *n*, and valve *m* makes a direct connection between *a* and *f* and *e* and *b*. The liquid, entering cylinder *g*, forces piston *h* down, which closes throttle valve *k* a corresponding amount, reducing the pressure of the entering steam. The valve stem is prolonged and attached to the end of lever *n*, and hence the downward movement of the throttle valve moves the relay valve *m* back to its original position. During this return movement of the

valve *m* the lever *n* pivots about its left hand end; while, when the lever was originally moved by the governor, it pivoted about its right hand end. Lever *n* is thus what is called a floating lever, the fulcrum of which is shifted from one end to the other, according to the conditions. When valve *m* has been returned to its original position, no further movement of the throttle valve can occur until the speed of the turbine changes again.

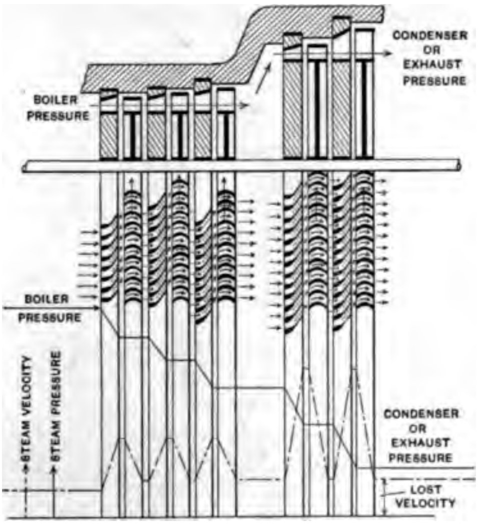


Fig. 12. Diagram of Hamilton-Holzwarth Turbine.

Hamilton-Holzwarth Turbine.

A turbine is being developed by the Hooven, Owens & Rentschler Company, Hamilton, Ohio, which is on the plan of the Rateau and Zoelly turbines, but differs in details of construction. In units of 750 Kw. and upward the turbine is divided into two parts, the high- and the low-pressure. Steam enters through a separator and passes through the main inlet valve and the regulating valve, all of which are below the bed plate. As the steam flows through the first set of stationary vanes it forms a complete ring instead of entering through a part of the circumference as in the Rateau turbine. The casing is divided into compartments with

one rotating wheel in each. Both the stationary and the moving vanes gradually increase in height toward the low-pressure end.

Fig. 12 shows the scheme of the turbine. This is somewhat misleading, in that the stationary nozzles apparently increase in area from inlet to outlet, a construction that would not be required with the small drop in pressure that occurs between the different compartments. The areas do not actually increase, however, because the guide vanes are so shaped that they are nearer together at the outlet side than at the inlet side, and to compensate for this

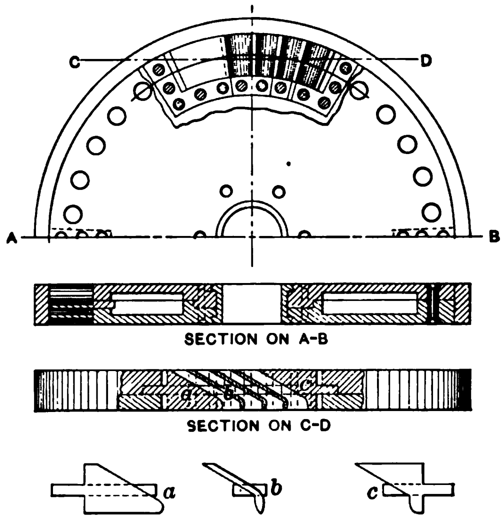


Fig. 13. Construction of Diaphragms.

they have to increase in height somewhat from the inlet to the outlet sides.

Details of Construction.—In Figs. 12, 13, and 14 are certain details of construction of the turbine. Fig. 13 shows the stationary discs which are built up of two side pieces riveted together. Each vane is a separate piece held by its projection at its lower end, which fits in an angular groove between the two disks at their periphery. The vanes are of drop-forged steel and are secured by rivets. After they are in position their outside ends are ground and a steel ring is shrunk on. In case it were not desired to extend

the vanes around the whole periphery the forms used at *A* and *C* would be employed as indicated in the section on *C D*.

The construction of the wheel is shown in Fig. 14. It is as light as possible with cast steel hubs, to which are riveted conical disks *A*. There is a space between the disks at their periphery in which are riveted their steel segments *B*. The vanes are attached to these

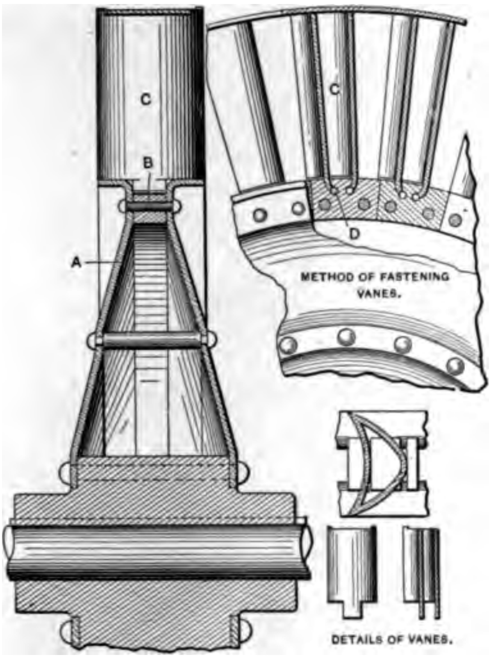


Fig. 14. Construction of Wheels.

segments, as shown in the sectional view at *D*. They are formed with lips extending downward and these lips are enlarged at their ends to fit into the enlarged bottom portions of cross channels in the segments *B*. The vanes are so designed that the passages through the walls have a uniform area from beginning to end and they are made hollow to reduce their weight. On the outer ends of the vanes a thin, steel band is shrunk to give an outside wall to the steam channels. The vanes are milled on both edges to give correct angles.

CHAPTER VI

COMPOUND IMPULSE TURBINES (Continued).

The Curtis Turbine.

The Curtis turbine is manufactured in this country by the General Electric Company, the large sizes at Schenectady, N. Y., and the small sizes at West Lynn, Mass. The Curtis marine turbine is being developed by a company headed by Mr. Curtis, the inventor, who is conducting extensive experiments.

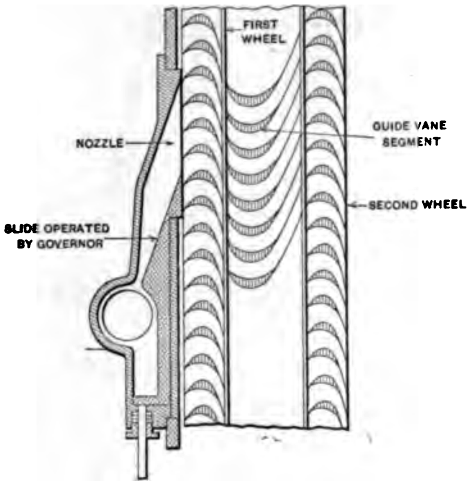


Fig. 1. Early Form of Curtis Turbine.

Early Type.—The machine is represented in its simplest and earliest form in Fig. 1. It consists of two rings of curved buckets mounted upon disks revolving with the shaft. Between the two revolving rings is a group of curved blades in the form of a short segment fixed to the interior of the turbine case. The nozzle is of rectangular cross section, so designed that one side of it can slide in or out without materially altering the ratio between the inlet and outlet areas of the nozzle. By this means the quantity of steam delivered is adjusted to suit the load, and it is not necessary to govern by throttling. An early turbine of substantially this

design, of 150 horse-power, was tested at Stevens Institute of Technology, Hoboken, N. J.

Stage Turbine.—In its practical form the nozzles are smaller in area than in the experimental machine mentioned and are arranged in groups; but the method of governing is in effect the same. One design that has been used is shown in Fig. 2. Steam

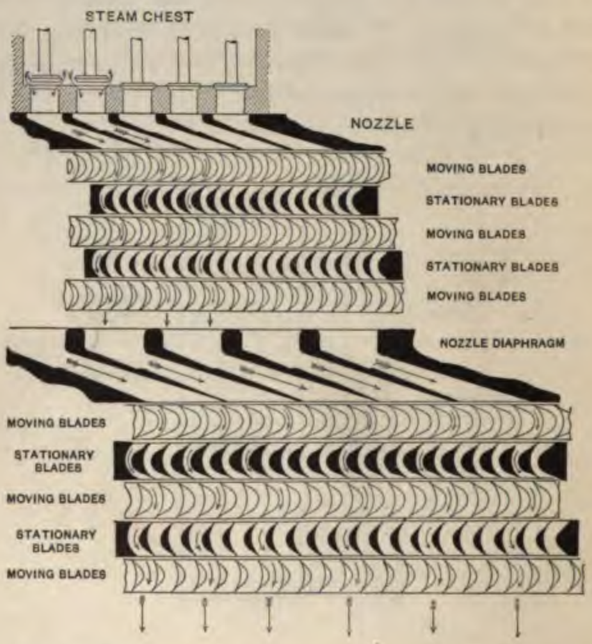


Fig. 2. Stage Turbine. Three Rotating Rings of Buckets in Each Stage.

enters through the series of nozzles, forming a broad belt of steam, and the quantity admitted is regulated by a series of poppet valves, one for each nozzle. Regulation is by opening or closing these valves automatically, which has the effect of increasing or decreasing the quantity of steam flowing, as may be required, without reducing the initial pressure. The turbine in Fig. 2 is a "stage" turbine, with two stages or elements, each consisting of three rotating sets of blades and the necessary guide vanes. Each element is incased in a separate compartment with its set of nozzles.

Since the earlier turbines were constructed, it has been found that better results can be obtained by dividing the turbine into more stages and using only two rotating rings of blades in each stage, and this plan is now followed in the larger sizes. Fig. 3 shows a half section of a turbine on this plan. The nozzle and its controlling valve are at the top, at the right, below which are the two sets of wheel blades, and the intermediate set of guide vanes.

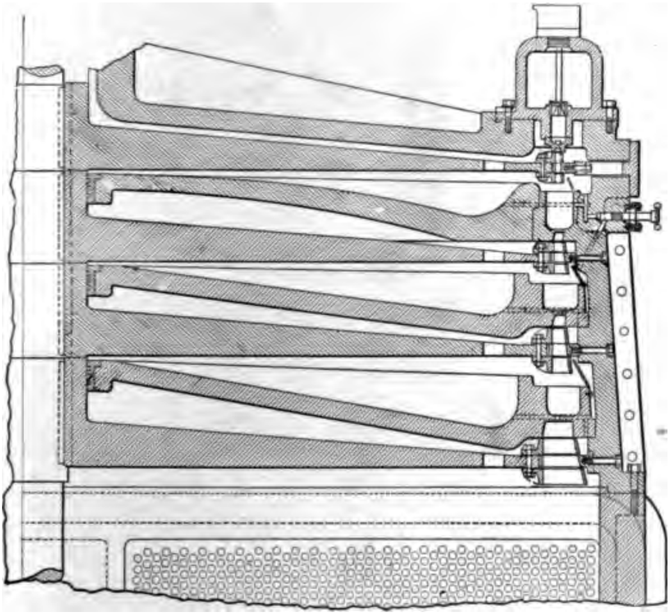


Fig. 3. Stage Turbine. Two Rotating Rings of Buckets in Each Stage.

Then follow in succession the second, third and fourth stages, with the wheel in each stage separated from the others by the diaphragms. Only the initial nozzles are controlled by the governor, but between the first and second stages are hand-operated valves, so that, should the pressure become too high in the first stage, steam may be delivered through these valves to the second stage. In later machines automatic spring-operated valves have taken the place of the hand valves.

Method of Reducing Rotative Speed.—The theoretical principles of the Curtis turbine have already been outlined in Chapter I. and

in the patent review of Chapter II., to which the reader is referred. Although a modified form of the De Laval expansion nozzle is used, the rotative speed of the wheel is much lower than in the

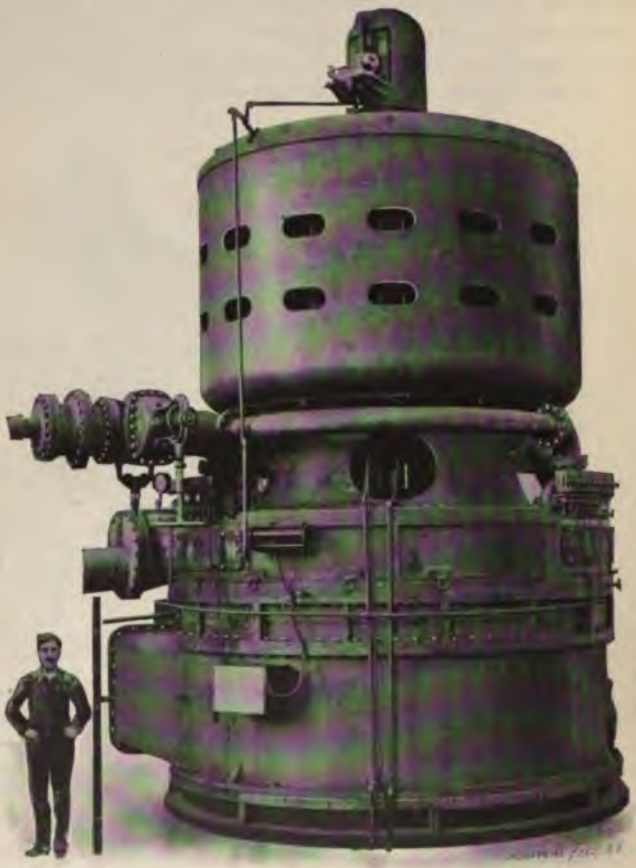


Fig. 4. The First 5,000 Kw. Turbine, Installed at the Commonwealth Station, Chicago.

De Laval type, since two or more rotating rings of blades are employed to utilize the high velocity of the steam after it leaves the nozzle. In the De Laval turbine the single wheel should run as nearly as practicable at half the velocity of the steam jet in order to absorb its energy. But in the Curtis turbine the speed is much

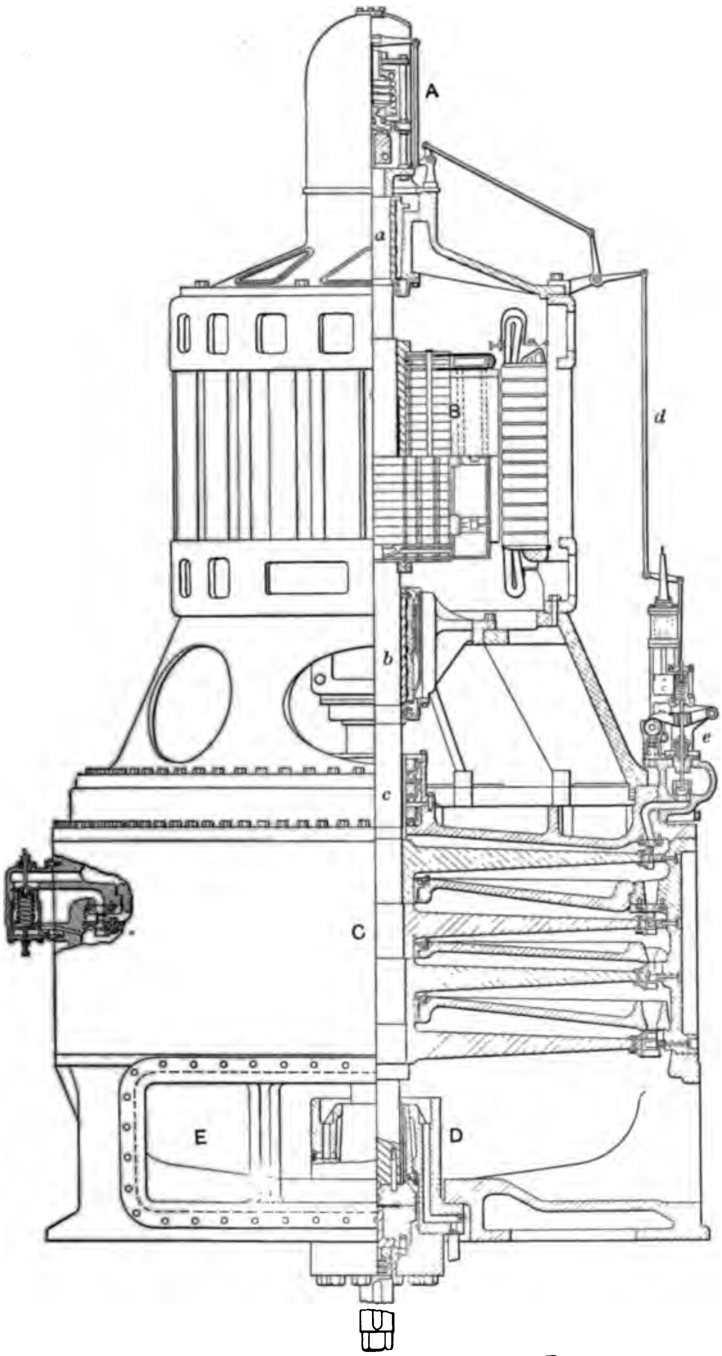


Fig 5. Sectional Elevation of 2,000 Kw. Curtis Turbine.

less than half the velocity of the steam, and when the steam issues from the first set of blades it has a high residual velocity; and this, in turn, is taken up in part by the second rotating set of blades, and so on. This construction makes it possible to utilize the energy of the steam with a comparatively small number of blades. For illustration, suppose steam to start with a velocity of 3,000 feet a second; once compounding would reduce the required velocity of the wheel by two, or to 750 feet per second instead of the 1,500 feet theoretically necessary with a single wheel, and three rotating sets of vanes would reduce the velocity to 500 feet a second.

Curtis Vertical Turbines.—The first commercial turbine built by the General Electric Company was a 600 Kw. unit, installed in their power plant at Schenectady in 1901. This machine was built on the lines advocated by Mr. Curtis, with a horizontal shaft and two stages with groups of wheels in separate casings, as in Fig. 2. Since the construction of this machine all the turbines of the 500 Kw. size and larger have been built with shafts in a vertical position, and the generator placed directly over the turbine. The total weight of the revolving parts is borne by a step bearing at the foot of the shaft, and the shaft is steadied and aligned by three bearings, one at the top of the generator, another near the foot of the shaft, and a third between the generator and the turbine. The sectional view, Fig. 5, shows the arrangement clearly. The different parts are lettered as follows:

A, spring-weighted governor; *B*, generator; *C*, casing inclosing the three turbine wheels; *D*, step bearing; *E*, outlet to condenser; *a*, upper steady bearing; *b*, lower steady bearing; *c*, stuffing box with graphite packing rings; *d*, connection from governor; *e*, mechanism operating admission valves; *f*, by-pass for maintaining correct pressure in second stage.

The considerations leading to the vertical design are stated by one of the engineers of the company, as follows:*

The relative positions of revolving and stationary parts are definitely fixed by the step-bearing. The stationary part is symmetrical, easily machined and free from distortions by heat. The shaft-bearings are re-

*W. L. R. Emmet, in a paper upon the "Steam Turbine in Modern Engineering," read before the American Society of Mechanical Engineers in 1904.

lieved from all strain, and friction is practically eliminated. The shaft is free from deflection and can be made of any size without reference to bearings, which can be placed where convenient and operated with surface speeds, which would not be practicable with the horizontal arrangement.

These features make possible the use of a very short shaft, and consequently the longitudinal spacing of moving and stationary parts is very little effected by temperature differences. The turbine structure affords

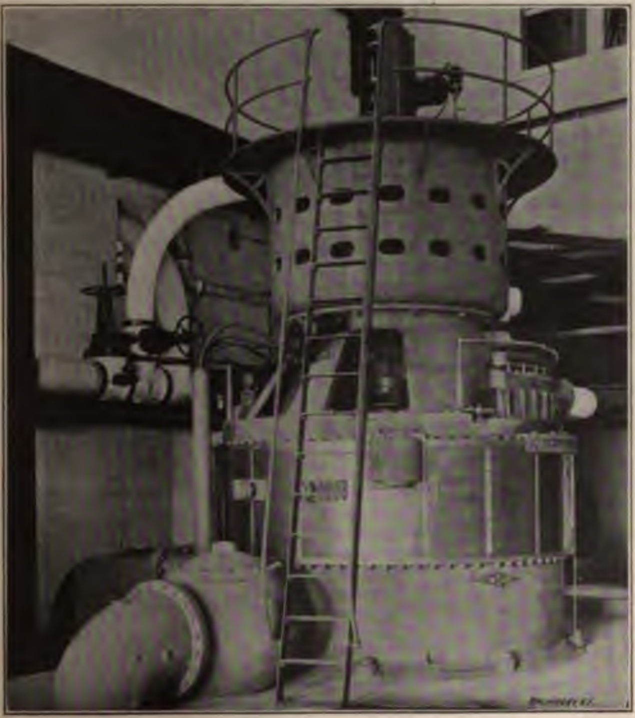


Fig. 6. 2,000 Kw. Curtis Turbine.

support and foundation for the generator. The cost of foundations is very small, and the solidity and alignment of foundation is not of vital importance. Much floor space is saved. All parts of the machine are conveniently accessible. Failure of lubrication cannot injure the shaft or other expensive parts.

Pressures and Velocities of the Steam.—In the four-stage turbine steam is expanded in the admission nozzles from an initial

Fig. 7. Bucket Segments—the Upper One for Low Pressure and the Lower One for High Pressure Sections of Turbine.

pressure of 150 pounds to 58½ pounds, thereby attaining a velocity of 2,000 feet per second. It acts upon the two rows of moving vanes and then, in passing through the second set of nozzles, is expanded to about 18½ pounds, again acquiring a velocity of about 2,000 feet per second. It here acts upon the second series of bucket wheels and is delivered to a third set of nozzles, which expand it to about 3½ pounds, imparting to it a velocity of about 1,600 feet per second. After acting upon the third set of wheels the process is repeated and the steam is delivered to a fourth set of nozzles, which expand the steam to about 1 pound absolute, giving it a velocity of 1,400 feet per second, which is absorbed by the fourth set of wheels, and by them the steam is delivered to the condenser with its energy practically all extracted.

Speeds of Rotation.—The speeds at which the various sizes of Curtis turbines (60-cycle) operate are as follows:

500 Kw.	1,800	revolutions	per	minute.
1,000 "	1,200	"	"	"
1,500 "	900	"	"	"
2,000 "	900	"	"	"
3,000 "	720	"	"	"
5,000 "	720	"	"	"

Fig. 8. Bucket Segment with Rim Riveted on.

smaller turbines the peripheral speed is about 400 feet per and in the larger ones it is reduced to 325 feet per second.

Turbine Buckets.—The most vital point in a steam turbine uckets, since they, and the spaces between them, must be correctly to give the proper direction of flow and the

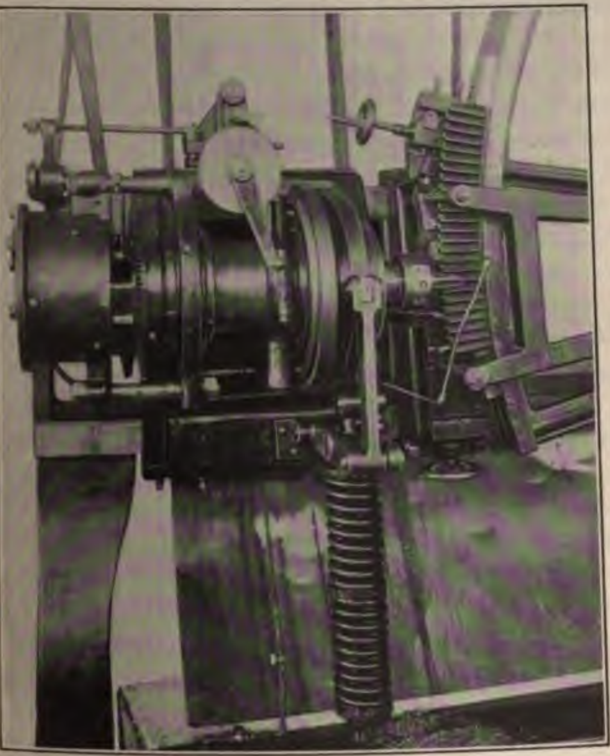


Fig. 9. Bucket Cutting Machine.

t mechanical efficiency, and also to provide for the progres- xpansion of the steam. The buckets of the Curtis turbine t out of the solid metal by special bucket cutting machines. smaller sizes the blades are cut from the disks comprising heels, and in the larger sizes the buckets are cut from seg- of steel or bronze and then bolted around the periphery of sks. In Figs. 7 and 8 are shown bucket segments, in the

first instance as they appear after machining and in the second with a rim of steel riveted on, closing the outer openings of the curved passages between the buckets.

Buckets are also made of drawn metal, the pieces being set in a mould and fixed in place by pouring molten bronze around them, thus forming one of the segments. In all these constructions the buckets themselves are less in width than the rim of the segment, so there is no possibility of their coming in running contact with any of the stationary parts of the machine.

While the process of cutting the buckets produces very nicely finished work, it is at best expensive, calling for special machines, which have taken a long time to design and develop. In all of them a single-pointed cutting tool is employed, the tool being so guided by the mechanism that its cutting edge will be in correct position for cutting effectively at all points of the curve. In older machines the tool was given a motion of rotation around the circumference of a circle (approximately, depending on the shape of the buckets), and as it passed the bucket segment it would remove a chip. The tool was gradually fed into the work as the cutting advanced. In the latest type the tool is given an oscillating motion, back and forth across the face of the segment. On the forward stroke the tool advances for the cut and on the return withdraws for clearance. The machine of this type is partly pneumatic in its action, and is an exceedingly interesting piece of mechanism.

Step Bearing.—In Fig. 10 is a sectional drawing of the step bearing. It consists of two cast-iron blocks, *A* and *B*, one carried by the end of the shaft and the other held firmly in a horizontal position and so arranged that it can be adjusted up and down by a powerful screw, *S*. Both blocks are recessed to about one half their diameter as shown at *C* and into this recess oil is forced through the central bore *D*, with sufficient pressure to raise the shaft slightly and support its weight on the thin film of oil which flows out between the flat faces of the two blocks. The lubricant flowing out fills the space surrounding these blocks and rises between the vertical bearing and the shaft, to the overflow *E*, where it escapes. The whole structure is inside the base and packing is used, aided by a low steam pressure, to insure that oil shall not escape into the vacuum chamber above. The pressure required in

the step bearing of a 5,000 Kw. machine is about 1,000 pounds per square inch and this is maintained by an electrically driven pump aided by an accumulator, so that if the pump should temporarily fail the pressure would still be maintained. There have been a number of instances where the oil pressure has failed and the lowering of the shaft and wheels after the flow of lubricant had stopped was at the rate of about .01 inch per minute. The

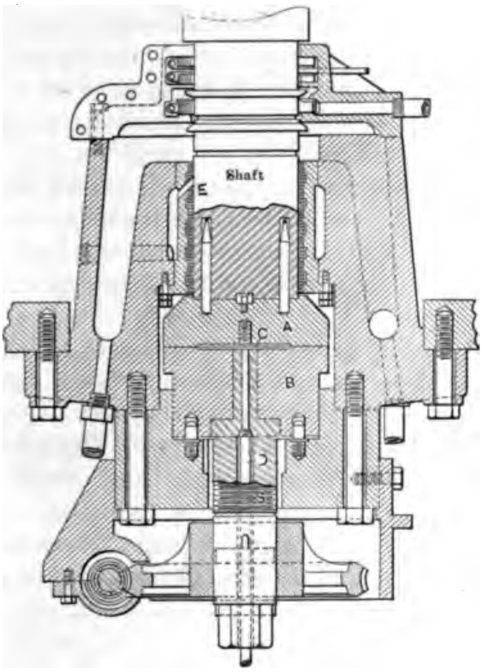


Fig. 10. Step Bearing.

wear on the blocks, however, did not appear to injure the wearing qualities of the bearing. In some later machines water has been used instead of oil to support the step bearing.

Small Turbines.—In a paper read before the American Street Railway Association in 1904, Richard H. Rice outlines some of the features of the small horizontal turbines built by the General Electric Company at their West Lynn works. Certain particulars of these are given in the accompanying table:

Rated Capacity. K. W.	Speed of Shaft. R. P. M.	Condensing or Non-condensing.	No. of Stages.	Current.	Poles.	Voltage.
1½	5,000	Non-cond.	1	Dir. Cur.	2	60
15	4,000–4,500	" "	1	" "	2	80-125
25	3,600	" "	1	" "	2	125-250
75	2,400	N-c. & Cond	2	" "	4	125-250
100	3,600	Cond.	3	Alt. Cur.	2	2,300
150	2,400	N-c. & Cond.	3 & 4	Dir. Cur	4	125-250
300	1,800	N-c. & Cond.	3 & 4	D. C. & A. C.	4	250, 500 & 2,300

The three smaller sizes have two bearings. The turbine wheels are overhung on the end of the shaft and the shaft is in one piece, with the turbine and armature both mounted on it. Beginning with the 75 Kw. size and upward the shafts are in two pieces and the sets have four bearings.

In the small sizes where the wheels are overhung the front end of the case may be taken off to obtain access to the wheels and intermediates, and in the larger sizes where four bearings are provided the upper half of the casing is removable for the same purpose.

In the four-bearing sets the generator and turbine shafts are united by a flexible coupling which permits some little inaccuracy in the alignment of the two shafts without affecting the operation of the set. This coupling is a modification of the Oldham coupling, the necessary flexibility being secured by the use of links turning on pins.

The 1½, 15 and 25 Kw. turbines are of the single-stage type, having a single group of nozzles and three rows of moving buckets. The larger sizes are multi-stage and have only two rows of moving buckets per stage.

The bearings used in these turbines are supported on spheres. ie linings are made in two parts and lubrication is effected by ced feed from a pump which is geared to the main shaft of the bine and supplies oil at a pressure of from three to six pounds r square inch.

Governing Mechanism.—Various methods of governing have en experimented with by the General Electric Company. In zes of 25 Kw. or less, governing is effected by throttling the eam pressure by the direct action of a powerful centrifugal govnor. In the larger sized machines, however, each nozzle or

group of nozzles is supplied with steam from a poppet valve operated by means of controlling mechanism under the influence of the governor. One method adopted for larger units consists in the use of a hydraulic cylinder with a controlling valve actuated by the governor. A movement of the controlling valve, caused by a change in the speed, admits oil to one side or the other of the piston of this cylinder and a movement of the cylinder results, through the intermediate mechanism, in the opening or closing of corresponding poppet valves. While the governor remains in any given position the hydraulic cylinder is also stationary and is locked in its position by confining the oil in both ends of the cylinder. A movement of the governor produces a corresponding movement of the hydraulic piston, and when this movement has taken place the parts come to rest. The motion of the hydraulic piston is transferred to a shaft running parallel with the bank of nozzles and on which is a series of cams that actuate the valves. The 2,000 Kw. turbine, Fig. 5, is controlled by a hydraulic gear of this type. The hydraulic cylinder is located in a vertical position above the nozzle valves, at the right, and its plunger moves the cam shaft one way or the other according to the position of the pilot valve. The cam shaft is plainly visible in the engraving. On some of the largest machines horizontal cylinders have been employed instead of the vertical, placed between the turbine and the generator, and with the plunger operating the cam shaft through a rack and pinion.

Mechanically Operated Gear.—This gear is a development from steam-engine practice and is used on some of the turbines manufactured at the Lynn plant of the General Electric Company. Each nozzle valve is actuated directly by a pair of reciprocating pawls, one adapted to open the valve and the other to close it. The several pairs of pawls are pivoted to a common moving support, which is oscillated by a rock shaft receiving its motion from the turbine shaft through a worm and wormwheel. At the upper end of the valve spindles are crossheads, in which are milled notches or teeth for the pawls to engage, and the engagement of the pawls in these teeth is determined by the angular position of shield plates controlled by the governor. These plates are set progressively, one in advance of the other, to obtain successive

actuation of the valves. When more steam is required, a shield plate permits the proper pawl to engage the crosshead of its valve and open the valve on the upward stroke; while if less steam is required the shield plates will be moved by the governor to such a position that the proper pawl will close its valve during the downward stroke of the rock shaft.

Another type of mechanical gear, that has been applied to smaller units, has positively actuated valves that are always either in the full open or entirely closed positions. Each valve has a crosshead and block and is actuated by a dog consisting of a small eccentric strap with a projecting arm about six inches long, provided with two hooks, one adapted to pull the crosshead block toward the eccentric shaft and open the valve and the other to push the block away from the shaft and close the valve. The governor controls the engagement of the hooks. The governing arrangement of this gear is very ingenious and sketches of the mechanism will be found in the 1906 report of the turbine committee of the National Electric Light Association.

Electric Governing.—One of the earliest methods used for controlling the nozzle valves, and which is still employed, is an electric system in which the action of the valves is governed by solenoids or magnets through which an electric current passes. In Fig. 11 is a diagram showing the principle of the arrangement. The governor at *G* connects with the cylinder *R*, on the surface of which is a series of contact points arranged spirally, so that as the cylinder turns one way or the other these points come in contact successively with corresponding points from which the vertical wires extend and close the circuit through these wires in succession.

Referring to the figure: *A* is the supply wire for the current and *B* the return. The current passes through the switch *S*, which ordinarily is closed, and thence to the wire and to the cylinder. The vertical wires at the left connect with the magnets belonging to the various sets of nozzles, but in this diagram the horizontal wires leading to one set of nozzles only are indicated, which accounts for several of the vertical wires having no apparent connections. When the cylinder *R* is so rotated by the governor as to bring two contact points together the current ener-

gizes the corresponding magnet at T, and thence passes to the return wire B. Should the turbine speed be above normal the governor arm drops and breaks the current at the switch S and all of the magnets are thrown out of action and their valves closed, shutting off steam.

The nozzle valves are not operated directly by the magnets, but through the medium of small auxiliary valves which the magnets

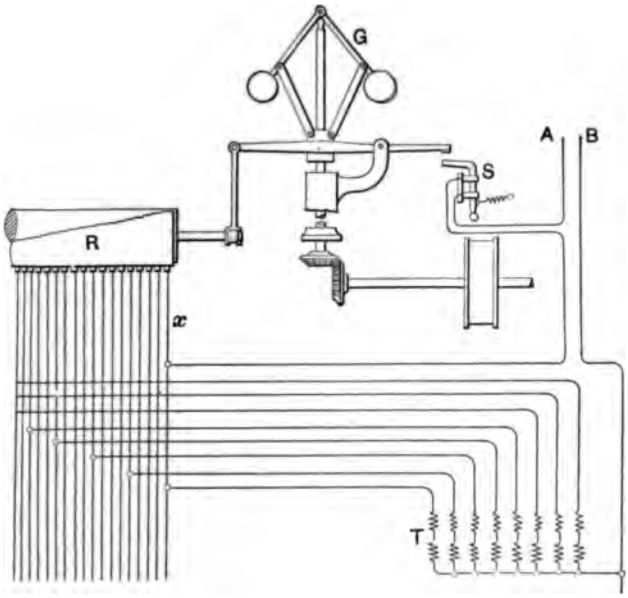


Fig. 11. Diagram of Electric Control for Nozzle Valves.

control and which serve to create a balanced or unbalanced steam pressure upon the faces of pistons attached to the nozzle valve spindles. One of the nozzle valves is shown in Fig. 12. The valve V is constantly pushed downward by the spring S, and the action of the valve is governed by the pressure in the space above the piston P, which is fastened to the valve stem A. Communication between the pilot valve and this space is had by the pipe I and when the pilot valve is in such a position that this pipe connection is open to the atmosphere the unbalanced pressure under valve V will be sufficient to raise the valve and allow steam to enter the nozzle. When, however, the pilot valve is in such a

position that steam under pressure is admitted to the space above piston P, this pressure, in connection with the spring S, forces the valve to its seat.

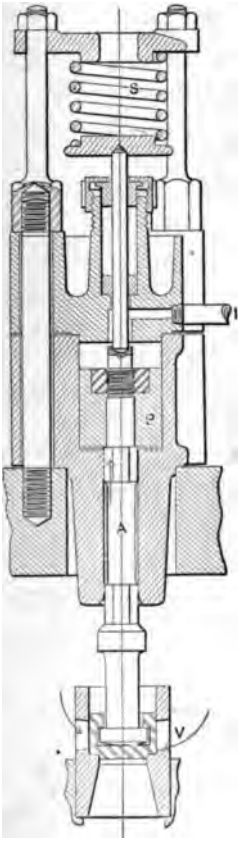


Fig. 12. Nozzle Valve.

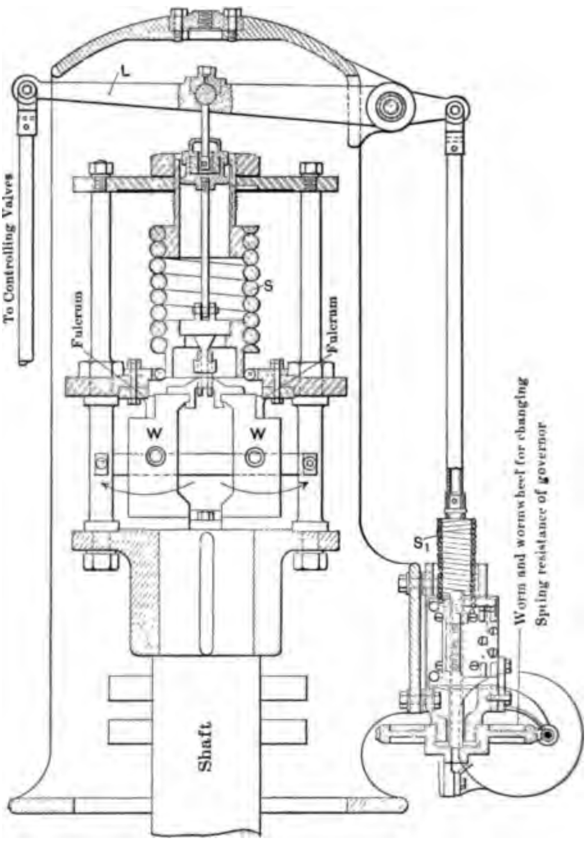


Fig. 13. Curtis Turbine Governor.

Curtis Turbine Governor.—In Fig. 13 is an outline of the governor for a 500 Kw. machine. The governor is supported by a flange keyed directly to the top of the vertical shaft of the turbine and the whole supporting framework rotates with the shaft. W W are the two weights fulcrumed at the points indicated and as the turbine speeds up the centrifugal force of the weights pulls the lever L downward against the resistance of the spring S.

A second spring at S_1 is arranged so that its tension can be increased or diminished to change the loading of the governor and thus bring the speed of the governor within small limits. The lever L connects with the valve mechanism.

Riedler-Stumpf Turbine.

This turbine has been developed and manufactured by the Allgemeine Electricitäts Gesellshaft, Berlin, who have now be-

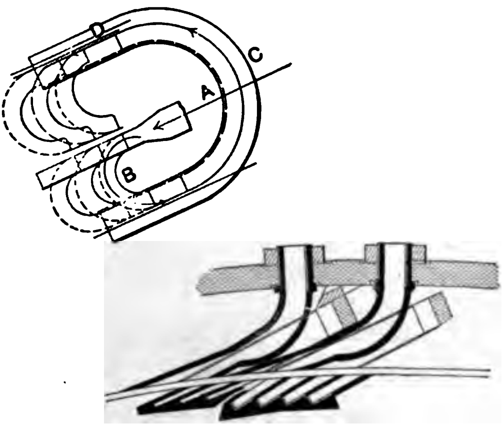


Fig. 14. Buckets and Guides of Compound Riedler-Stumpf Turbine.

come incorporated with a new Berlin organization, known as the Union Electric Company. The object of the Union Company is to exploit in certain European countries important steam turbine patents, chiefly those of Professors Riedler and Stumpf, controlled by the Allgemeine Company; and the Curtis patents, owned by the General Electric Company in this country. As a result of this organization, turbines are now being constructed by the A. E. G., combining features to be found in both the Curtis and the Riedler-Stumpf machines.

The Riedler-Stumpf Compound Turbines.—The single wheel Riedler-Stumpf turbines have already been described in Chapter IV. In the compound turbine of this design, in which two or more wheels, or else two or more rows of buckets on the same

wheel, are used, each wheel is provided with semi-circular buckets. The steam is projected against one side of the buckets of the first wheel and then as it escapes from the opposite side, it is collected by curved guides which carry it around to the next wheel. The sketch, Fig. 14, shows the arrangement, and it will be noted that here it is not necessary to arrange the guides spirally in order to

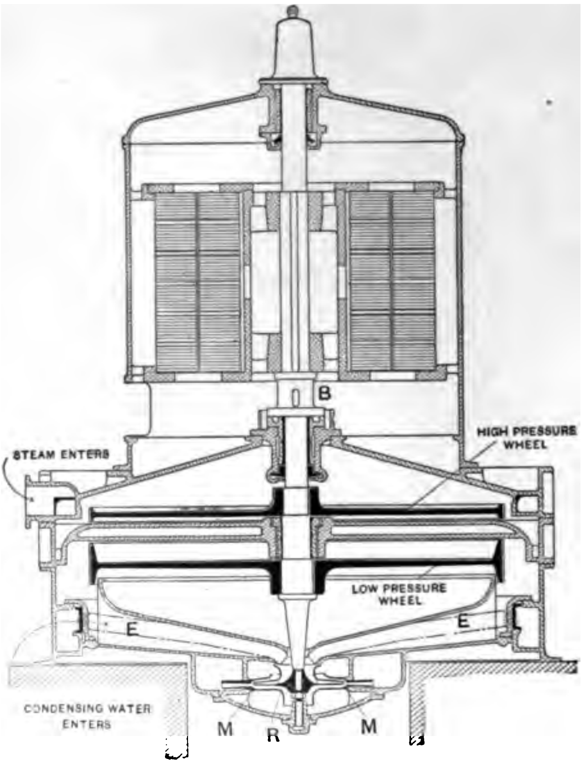
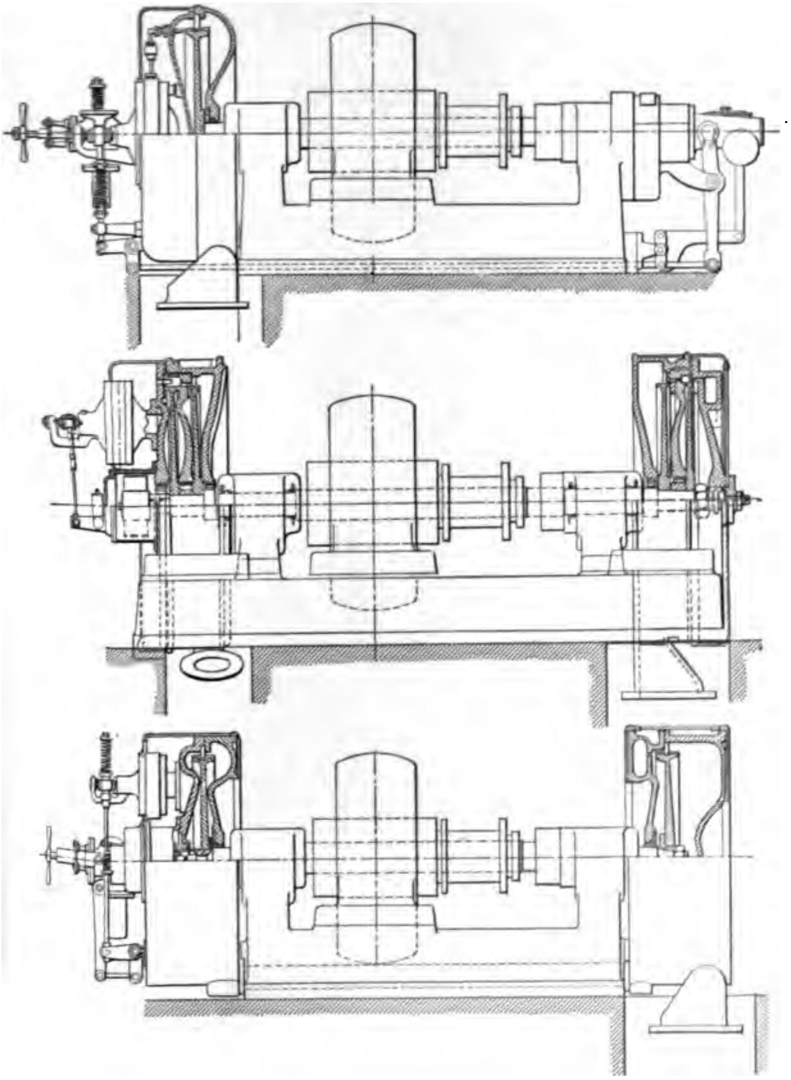


Fig. 15. Compound Riedler-Stumpf Turbine.

circumvent the nozzle as was done in the case of compounding with the single wheel as described in Chapter IV. The letters *A*, *B*, *C*, *D* indicate the direction of the flow of steam in Fig. 14. When a considerable speed reduction is desired, the turbine is divided into stages with two steps in each stage.

In Fig. 15 is a sectional drawing of a two-stage turbine, similar in its arrangement to the Curtis turbine made in this country. The



Figs. 16, 17, and 18. A. E. G. Turbines.

upper wheel may be classed as a high-pressure wheel and the lower a low-pressure wheel. Steam enters as indicated, and is conducted by suitable passages from the first to the second wheel. It then enters the annular passage *E E*, where it comes in contact

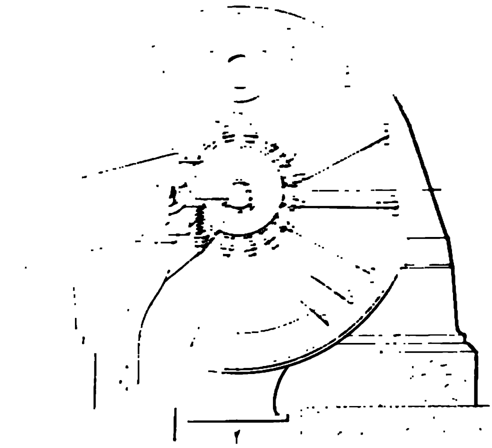


Fig. 19 End View of A. E. G. Turbine.

The A. E. G. Turbine.—Figs. 16, 17 and 18 show three of the more recent designs of the Allgemeine Company, in two of which the principle of the Curtis type of wheel is utilized. The simplest design is [illegible] turbine, the wheels of which are made [illegible] In Fig. 17 is a four-stage turbine [illegible] stage; while in Fig. 18 is a [illegible] In each of these designs [illegible] by the ends of the shaft which [illegible]rhangs, while the generator is [illegible]

[illegible] the A. E. G. T[illegible]—The use of [illegible]ng unit simplifies the construction

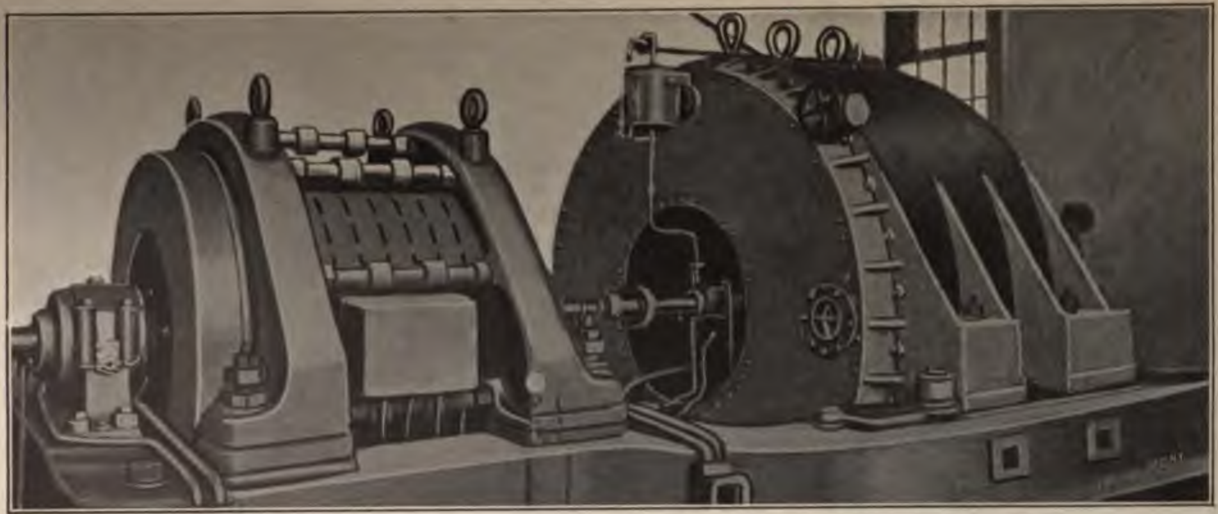


Fig. 20. 2,000 H. P. Riedler-Stumpf Horizontal Turbine and Generator.

to a marked degree, and with a very stiff frame and a heavy shaft, gives satisfaction. Owing to the stiffness of the frame it is possible to secure the casing to it so that it is possible to expand and contract. The wheel casings are of cast iron provided with relief valves as a protection against a possible rise of pressure. The wheel is held on the shaft end by a flange and is machined out of a solid nickel steel disk. The steam inlet to the main cut-off valve and to the steam distributing chest is through a fine-meshed screen. The steam from the distributing chest is delivered to the nozzles by a number of pipes shown in end view, Fig. 19. The bearings are supplied with oil under pressure and are lined with white metal. The governor is fitted direct on the free end of the shaft and is of the spring type. It is placed inside the steam distribution chest and openings which lead from the latter to the nozzles are closed or opened by a steel band.

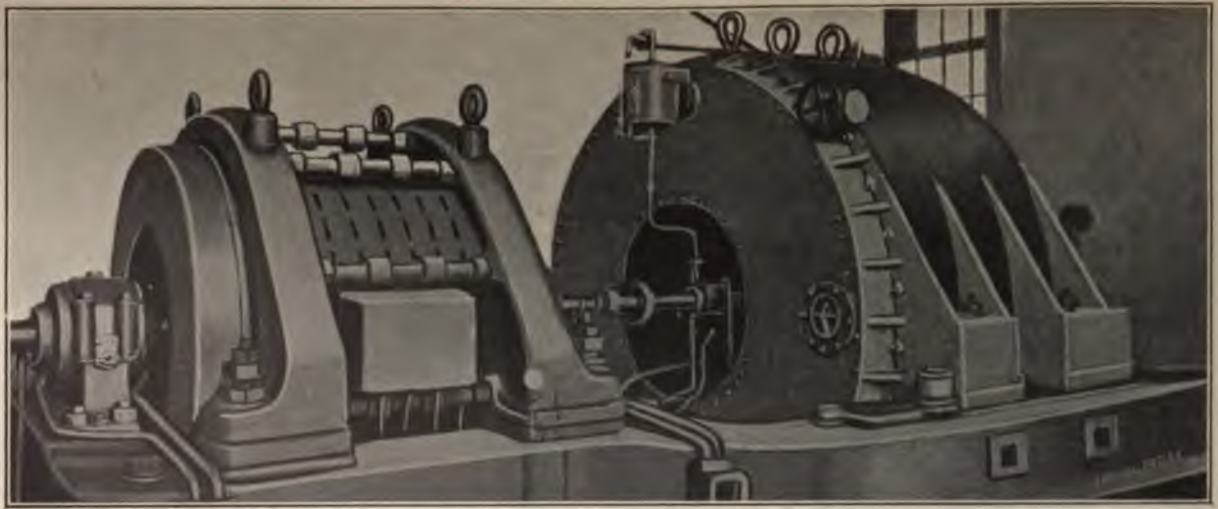


Fig. 20. 2,000 H. P. Riedler-Stumpf Horizontal Turbine and Generator.

General Principles.—In the Parsons turbine there are alternate rows of stationary guide vanes and moving wheel vanes as in Fig. 1. The steam flows through a fixed ring of directing blades, which serve the purpose of steam nozzles, onto a revolving ring of similar blades and so on, the pressure being reduced a small amount at each step. The revolving rings of blades act both in the capacity of buckets and nozzles as in any reaction turbine.

Assume, for illustration, that the steam expands from 115 pounds absolute to atmospheric pressure in its passage through the turbine and that there are 40 rows of guides and vanes giving an average drop in pressure of 2½ pounds at each wheel. If steam were to flow through an expanding nozzle from 115 pounds to 15 pounds absolute, its velocity would be about 2,700 feet per second; but, by stepping down the pressure and allowing it to expand an average of 2½ pounds at each stage, the velocity of flow corresponding to the differences of pressure would be only about 400 feet per second. If the Parsons turbine were purely a reaction wheel, the wheel would travel nearly as fast as the steam when it left the moving vanes, and in the above illustration would have a peripheral speed of nearly 400 feet per second. In the actual turbine the average speed is much less than this, requiring more rows of blades and an immense number of blades. In a 400 Kw. turbine there are 58 rows of guide vanes and wheel vanes, or 116 rows in all, aggregating about 30,000 blades.

The pressure differences at each element or set of blades gradually decrease from inlet to exhaust, instead of running uniform as in the above example, since, for a given difference of pressure, the velocity of steam is much greater at low than at high pressures. Thus, in flowing from 165 pounds to 155 pounds absolute, a difference of 10 pounds, the velocity acquired is only about 520 feet per second; while at atmospheric pressure practically the same velocity is acquired by a drop of only one pound in pressure or 1/10 as much as in the first case. The steam velocities are kept within 150 feet per second as a minimum at the high-pressure end, and 600 feet per second as a maximum at the low-pressure end.

Westinghouse-Parsons Turbines.

Description of Parts.—The elemental parts of a Parsons turbine are the rotor or rotating element, the stator, comprising the casing and guide vanes, and the balancing pistons. These are shown in Fig. 2, which represents a Westinghouse-Parsons turbine. Steam enters the chamber *B* at boiler pressure through the steam pipe *A* and passes to the right through the first group of blades which gradually increase in height (see Fig. 11) to chamber *C*. Here, to avoid excessively long blades as well as many sizes of blades, it

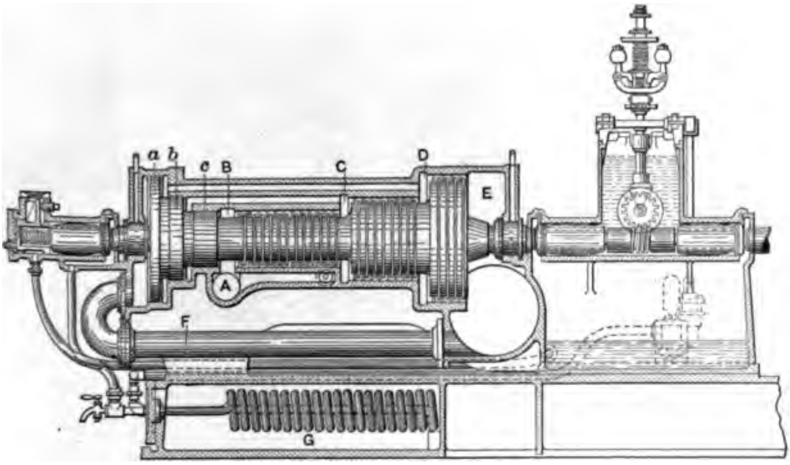


Fig. 2. Sectional Elevation of Westinghouse-Parsons Turbine.

is necessary to jump to a larger diameter and the steam flows through a second set to *D* and finally through a third set to space *E*. The balancing pistons *a, b* and *c* are of such a diameter that the steam pressure against them exactly balances the axial thrust in the direction of the steam flow. This thrust is composed of three factors: (1) The static pressure on the end of the drum; (2) the forward thrust on the blades due to the impact of the steam; and (3) the backward thrust due to the reaction of the steam in leaving the blades. The net result is a forward thrust. The diameters of the pistons are approximately equal to the mean diameters of the steam areas of the different steps. The pipe *F* connects the space back of the balancing pistons with the exhaust chamber. *G* is a coil for cooling the oil circulating through the bearings.

In Fig. 3 is shown a 400 Kw. turbine open for inspection. The casing is made in halves divided longitudinally so that the upper half can be removed, exposing the rotor, which may then be raised from its bearings, after the bearing caps are removed. The interior walls of the casing contain the stationary radial blades corresponding to those on the rotating cylinder. Starting at the left, it will be seen that there are several rows of blades, all of the same height; then there is a change to blades of a slightly

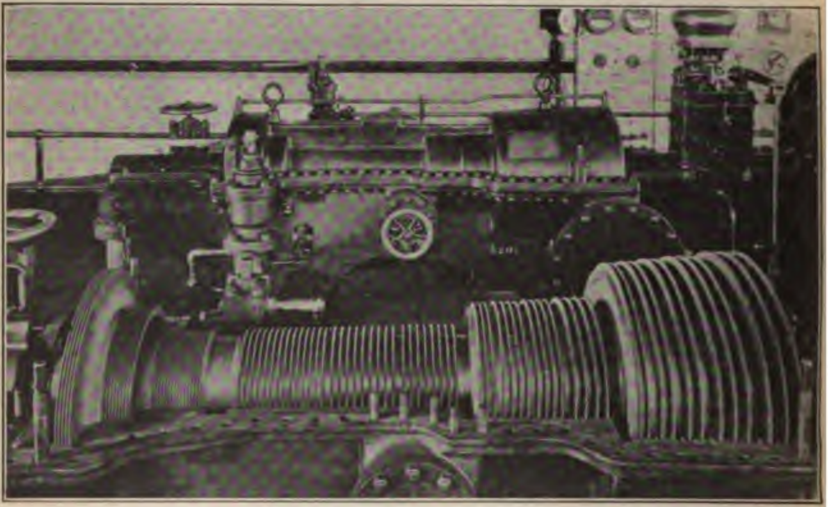


Fig. 3. 400 Kw. Westinghouse-Parsons Turbine with Casing Removed.

greater height and there are several rows of this size, and so on. When the mechanical limit is reached for size of blade, the rotor is then increased in diameter, giving a greater circumference and allowing shorter blades. The correct method, theoretically, would be for each row to be a little higher than the previous one throughout the turbine. Practically, this is neither convenient nor necessary.

A more detailed description will now be given of certain parts of the turbine, as made by the Westinghouse Machine Company.

Turbine Blades.—These were formerly made of a special cold-drawn bronze, but at the present time drawn steel is extensively used. The blades are secured in annular rings turned on the out-

side of the rotor and recessed on the inside of the casing, by calking after they are in place. The blading material is drawn out into long strips and sawed up to the proper blade length. The blades are separated by soft steel distance pieces which just fill the grooves between them and maintain the proper entrance angle. For the extreme low-pressure blading of the large turbines drop-forged steel blades are employed, with separators forged directly on the base of the blades. For maintaining uniform spacing between the outer ends of the larger blades and also to stiffen them, a flat steel lace is threaded through openings near the outer edge

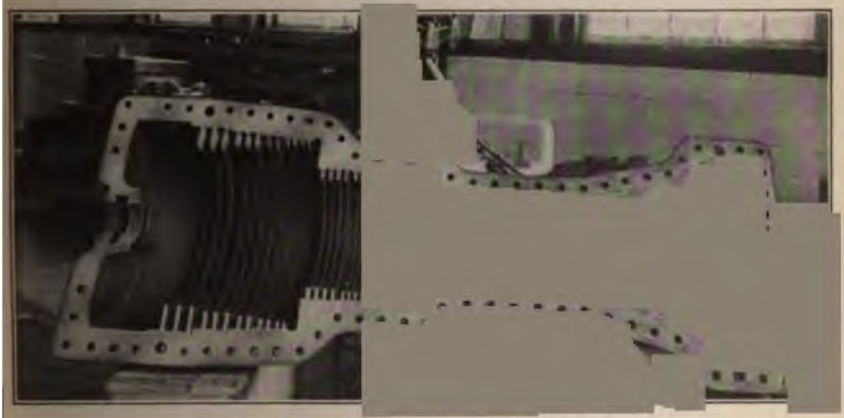


Fig. 4. Upper Half of Casing of 400 Kw. Turbine.

of the blades, and twisted between the adjacent blades. The smaller blades have ⅛-inch clearance sidewise and the larger from ½ to 1 inch, so that dangers of colliding are remote; and if this occurs the turbine may still be operated.

Bearings.—When the Parsons turbine was first experimented with trial shafts were run with bearings of different descriptions up to very high velocities, and it was found that no difficulty was experienced, provided the bearings were designed to have a certain amount of give or elasticity. It was then determined that elasticity combined with frictional resistance to the transverse motion of the bearings gave the best results and led to the adoption of the bearings shown in Chapter II. in connection with the Parsons patents. The shell was surrounded by a series

of friction rings, each being alternately larger and smaller than the adjacent ring, the small series fitting the shell on the outside and the large series fitting the hole in the bearing block.

In the Westinghouse-Parsons turbine the bearings are made up of several concentric sleeves instead of the rings loosely fitted in the pedestals. Oil circulates between the sleeves, and the capillary action forms a fluid cushion about the several sleeves, which restrains vibration and at the same time gives sufficient flexibility to allow the shaft to revolve about its axis of gravity instead of its geometrical axis. The bearing proper is a gun

Fig. 3. Ribbed Disk for Water-packed Gland.

metal bushing which is prevented from turning by a loosely fitted dowel. Outside of this are three other concentric tubes.

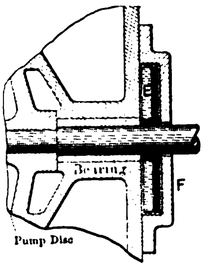
Water-Packed Glands.—In any turbine it is necessary to provide glands at the ends of the casing to prevent the escape of
...dmission of air around the shaft, which latter is
...e case of high vacuum. Steam-packed glands
...ith success, but the Westinghouse company now
... glands. In the casing is an annular groove
... in which rotates freely a disk attached to the
... has vanes on its faces, like the blades of a
...ig. 3.) The compartment is filled with water,
...rbine is running the water is thrown outward
... fills the outer part of the annular space and
...or steam from passing the periphery of the disk.

A similar device has been employed in the glands of centrifugal pumps.*

Lubrication—A small pump, driven by a worm and wormwheel upon the shaft circulates oil through a closed system, comprising in the order of arrangements, pump, oil cooler, bearings and reservoir. The oil is supplied to the bearings at the top, at one :nd where it follows a groove in the top of the shell, from which t is distributed around the shaft. A forced circulation under ııgh pressure is not found to be necessary. The object is to naintain an oil film around the journals so they will never actually ome in contact with the bearings.

Governing Arrangement.

Description of Governor.—Fig. 6 shows the governing mechanm of the Westinghouse-Parsons turbine. It is substantially ıe same as that illustrated in the second chapter in connection ith the Parsons patent of 1896. The governor is of the centrifgal type with bell-crank levers, the vertical arms of which carry ıe balls, and the horizontal arms bear against the spiral spring, hich resists the centrifugal force of the balls. The tension of ıe spring may be adjusted for the purpose of synchronizing two ternating current generators when running in parallel. The ain admission valve is actuated by the piston *B*, which is conolled by the pilot valve *A*. Steam is admitted below the piston ırough the annular clearance around the main valve stem and ıises the piston against the pressure of the spring. When the ilot valve *A*, however, uncovers one of the ports the steam



*A simple device for packing the shaft of centrifugal pumps a frictionless manner was introduced some years since by essrs. Robinson Brothers & Co., of Melbourne, Australia, d described in a recent issue of London *Engineering*. It applicable as well to the packing of the shafts of steam rbines, where the efficiency is impaired by the leakage of : into the vacuum end. The idea is ingenious in its very nplicity. The bearing is cased in with an annular chamr *F* which is filled with water. Rotary motion is given to e water by means of a disk or set of vanes *E* attached to e shaft. As the disk is tight upon the shaft, any air to ach the interior must pass around the tops of the blades. ıe water in which the tips are immersed, however, is under eater pressure, due to centrifugal force, than the pressure the atmosphere can overcome, and thus the air is effectıly excluded without the entrance or expenditure of water while the shaft is left ıtirely free.—*Power*, April, 1904.

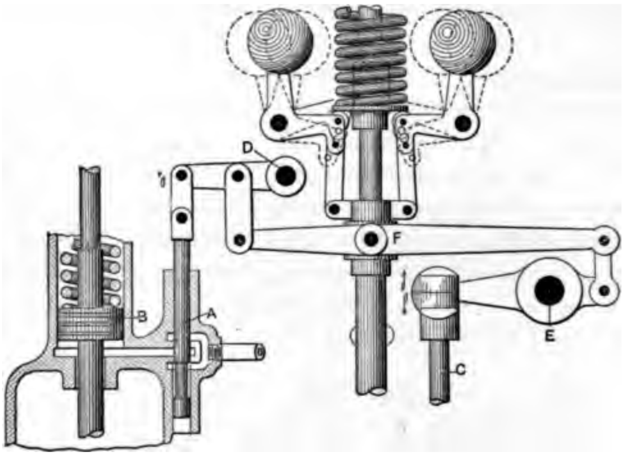


Fig. 6. Governing Arrangement.

escapes from the space under the piston, through the small exhaust pipe and allows the spring to close the valve.

The pilot valve is governed both by the motion of the governor and the reciprocating motion of the rod *C* which is actuated by an eccentric driven through a worm and wormwheel from the main shaft of the turbine. *D* and *E* are fixed fulcrums and *F* is a floating fulcrum moving up and down with the governor sleeve. The reciprocating motion of the rod *C* is communicated to the pilot valve *A* and thence to the main valve, admitting steam to the turbine in puffs; while the distance that the valve

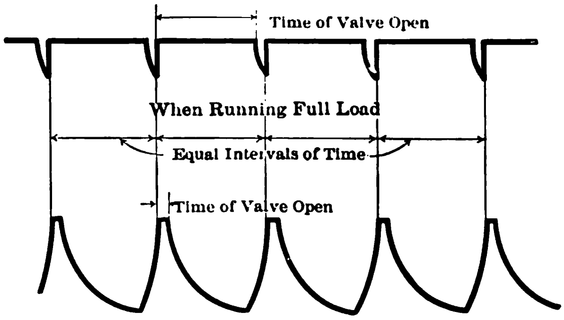


Fig. 7. Indicator Diagram Showing Effect of Reciprocating Valve.

opens is determined by the position of the governor. The pilot valve was also originally designed to act as a safety stop, but an auxiliary and entirely separate safety stop is now used for actuating a quick-closing throttle valve. Fig. 7 shows an indicator card taken on a Parsons turbine at two different loads, the indicator being attached to the admission space A in Fig. 2. The indicator barrel was revolved at a constant speed. At light loads the valve opens for very short periods and remains closed

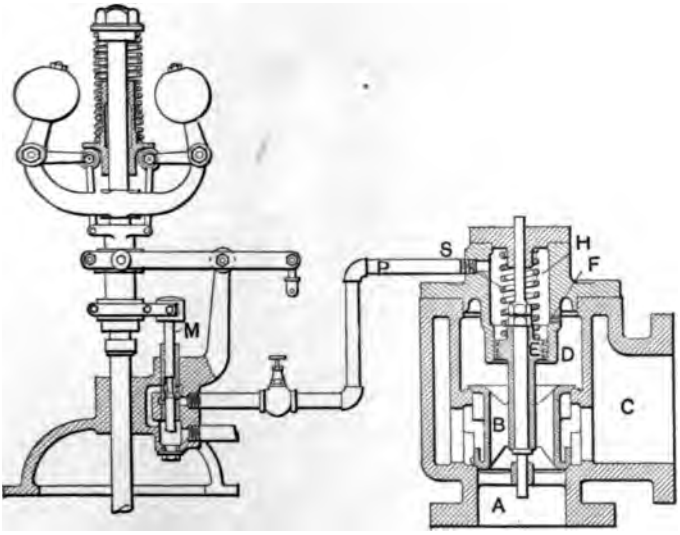


Fig. 8. Valve for Admitting High-Pressure Steam to Low-Pressure End of Turbine.

for the greater part of the interval. As the load increases the valve remains open longer, until finally almost continuous pressure is maintained in the high-pressure end of the turbine. The net effect of this method of governing is the same as though steam were throttled in the usual manner. While it is true that by admitting steam in puffs high-pressure steam is used at all loads at each oscillation of the valve, as has been claimed for the device, there is also excessive throttling of the pressure at each puff. This arrangement has the advantage, however, of removing the possibility of the valve sticking, since it has a continuous motion.

Auxiliary Governing Valve.—A secondary admission valve is

provided to admit high-pressure steam to the second drum of the turbine on overloads and increase its capacity up to 50 per cent or more in excess of the normal rating. This arrangement has the further advantage that it enables much better economy to be maintained under normal loading than when the primary admission valve only is used for governing. In Figs. 8 and 9 are diagrams showing the principle of the valve, and its connections, taken from the patent records, and the sectional view, Fig. 11, shows the actual arrangement of both primary and secondary valves.

In Fig. 8 *A* is the steam inlet and *B* is a valve admitting high-pressure steam, when the valve is raised, to the port *C*, which

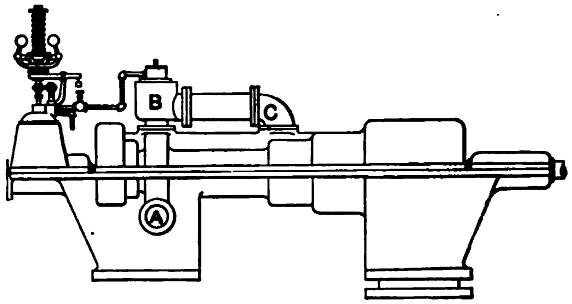


Fig. 9. Showing Connections for By-Pass.

connects with the intermediate part of the turbine. This is shown in Fig. 9, where steam enters at *A*; *B* is the valve, and *C* is the pipe leading to the turbine.

The valve, *B*, is hollow, allowing the steam to pass through to the space, *D*, where it bears against the under side of the piston, *E*. *F* is a small passage leading from space, *D*, to the space, *H*, so that under ordinary conditions there will be a balanced pressure on the piston *E*, and the valve will be kept seated by the spring, *S*. Connecting with the space, *H*, in which the spring is located, is a pipe, *P*, leading to a by-pass in the base of the governor, shown at the left, which is opened or closed by a pilot valve, *M*, under control of the governor. Under normal conditions the pilot valve will be in the position shown, closing the by-pass and preventing the escape of steam from the chamber, *H*. Should the speed of the engine decrease beyond a fixed point, however,

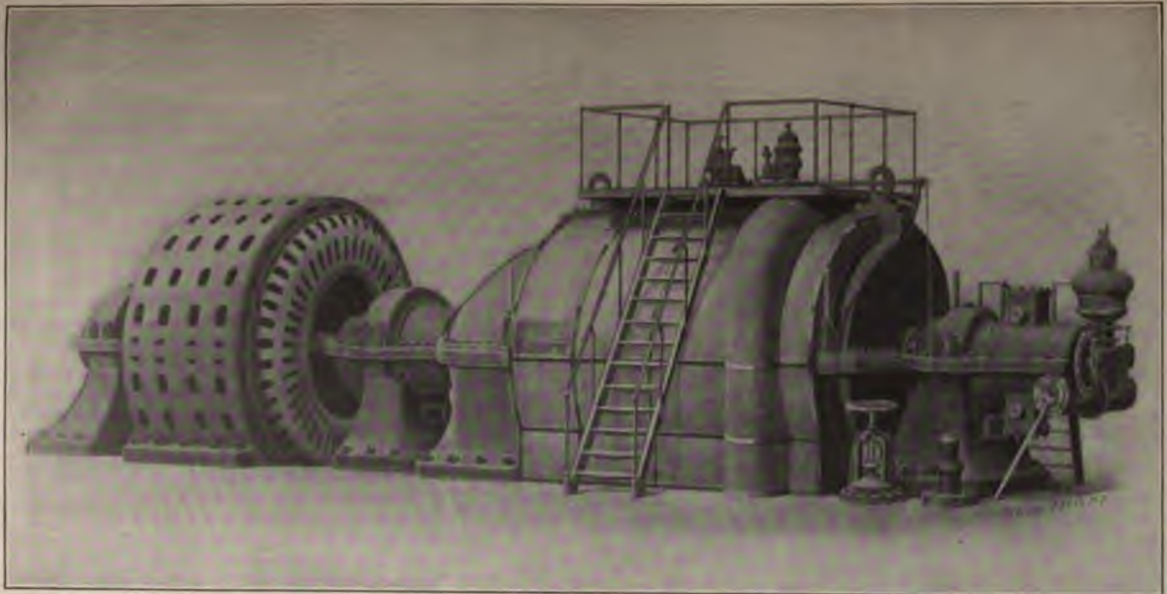


Fig. 10. 5,000 Kw. Westinghouse-Parsons Turbine.

the governor balls would move inward, which would depress the governor yoke and the pilot valve, *M*, causing the pilot to open and allow steam to escape from space, *H*, into the atmosphere. The result would be an unbalanced pressure on the piston, *E*, causing it to raise and compress the spring and open the valve, *B*, allowing the high-pressure steam to enter the low-pressure part of the turbine and increase the power.

Sectional View of Westinghouse-Parsons Turbine.—In Fig. 11 on the opposite page is a sectional view showing the essential parts of a turbine, which are lettered to correspond with the following list of parts:

S.—Steam admission.

V.—Admission valve to high-pressure end. This valve is controlled by the governor (connections not shown) and is oscillated by an eccentric driven by the worm and wormwheel at right-hand end of main shaft.

V_s.—Auxiliary valve also controlled by governor and opened automatically in case of overload.

P, P, P.—Pistons or disks against which steam pressure acts to balance the thrust.

E, E, E.—Equalizing pipes. The two upper ones maintain steam pressures against the front faces of balance pistons equal to the pressures in the steps or stages of the turbine having corresponding diameters—the lower one maintains vacuum pressure at the back of the large piston.

T.—Adjustment bearing for maintaining the exact running position of the rotor and for taking up any unbalanced thrust not provided for by the pistons. This bearing is adjusted endwise to locate the rotor in correct position relative to the stationary part.

R.—Relief valve.

B.—Exhaust passage.

The arrangement of the governor has previously been described.

5,500 Kw. Turbines.—The general view, Fig. 10, shows a turbine of this power of the type that is to be used for the initial equipment of the Pennsylvania Railroad terminal property in New York City, furnishing electric power for the trains passing through the tunnel approaches to New York City now in construction. The space occupied by this machine is approximately 27 feet 8 inches by 13 feet 3 inches, and the height to the top of the rail is 12 feet. It occupies less than $\frac{1}{20}$ square foot per electric horsepower and develops 20.2 horse-power per square foot of floor area. For the complete unit, including the generator, a space of the above width and 47 feet 4 inches long is required.

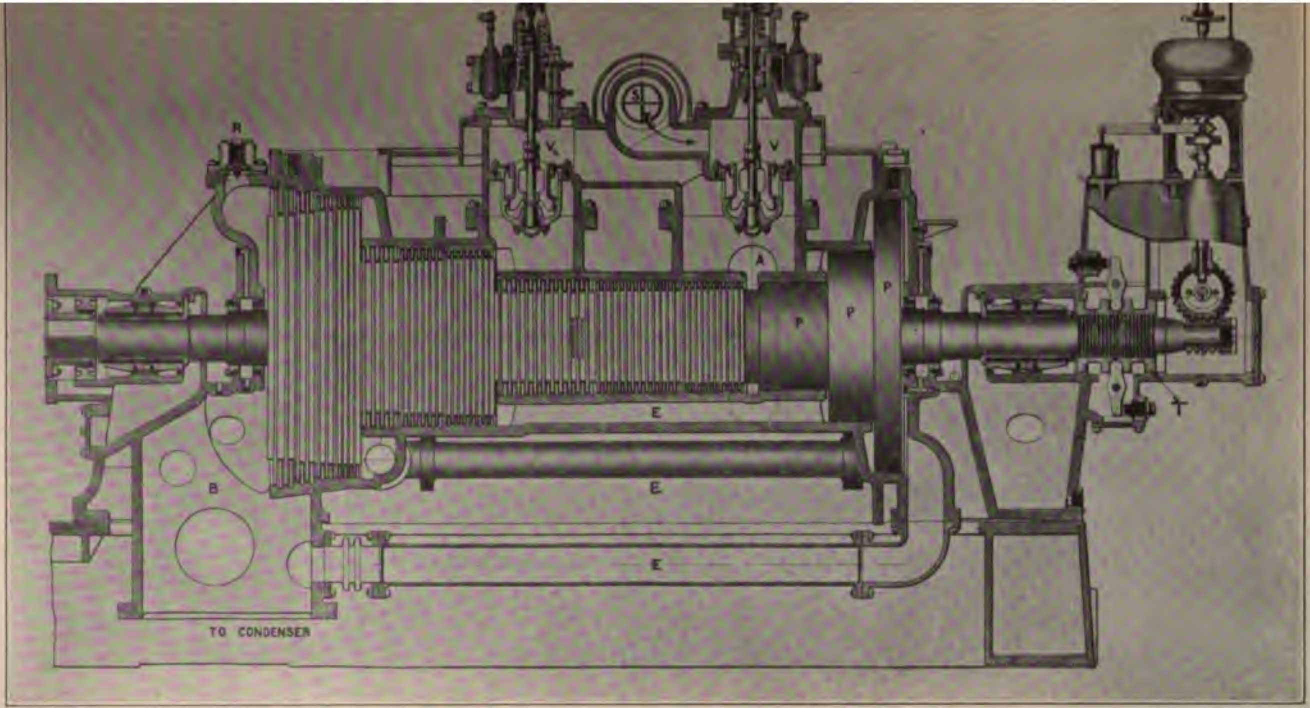


Fig. 11. Sectional Diagram of Westinghouse-Parsons Turbine.

The machine runs at 750 revolutions per minute. The construction is substantially that used in the smaller machines. To support the drum a central steel quill is employed. Hollow forged steel ends are forced into the ends of the quill and constitute the journals. High-pressure steam is conveyed to all parts of the quill structure to eliminate distortion due to expansion.

The bearings of these larger machines are of the self-aligning type, similar to those employed in generators and cross-compound engines. The departure from the oil cushion journals used in the smaller machines is made possible by the low speed of rotation.

Turbines with Separate High- and Low-Pressure Cylinders.—Three turbines, aggregating 3,750 Kw., have been installed in the immense power plant of the Interborough Rapid Transit Company, New York City, to furnish current for lighting the New York subway. The most striking feature of these is the separation of the high-pressure and low-pressure sections, which provides for a central bearing for the turbine shaft, thus reducing the distance between bearings and allowing a much lighter shaft. As originally designed, these turbines were to be equipped with reheaters, placed between the cylinders, but the reheaters were finally omitted. It was expected that the drying of the steam and possibly its superheating in the receiver, before the steam entered the low-pressure cylinder, would effect an improvement in economy. Experiments at the Westinghouse shops, however, have demonstrated that such is not the case. Francis Hodgkinson states that "exhaustive tests have shown the reheater to be of little, if any, value in increasing the economy of the turbine when the high-pressure steam condensed in the reheater coils was charged up against the turbine. An improvement in the separator resulted in an improvement in the operation of the reheater, but, notwithstanding this, no advantage due to the reheater could be observed and its application does not seem to be warranted, on account of the decreased compactness of the machine."

Westinghouse-Parsons turbines are no longer built with two cylinders, the whole tendency being toward compactness. The 1,500 Kw. unit with inclosed generator, Fig. 12, is of the latest type. The objects of encasing the generator are to reduce the noise which sometimes is enough of a roar to be disagreeable, and

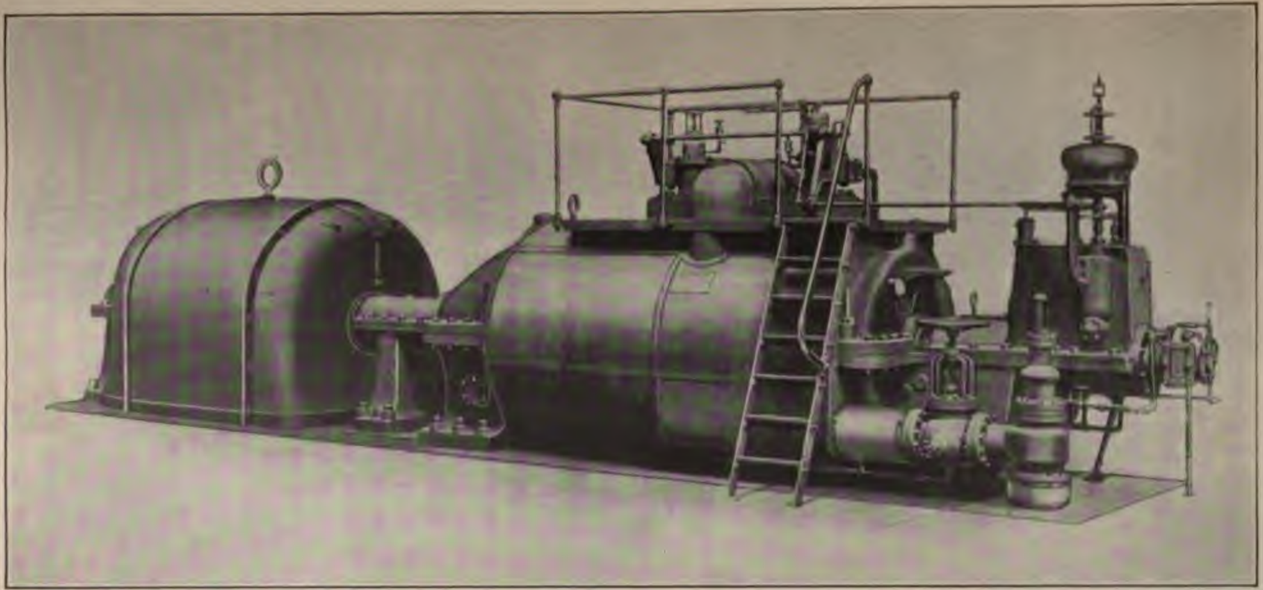


Fig. 12. **1,500 Kw. Turbine Unit with Inclosed Generator.**

to enable a current of air to be forced through the generator for the purpose of maintaining a moderate temperature and enabling heavier overloads to be carried without danger of overheating.

The Brown-Boveri Turbine.

Description.—In Fig. 13 is shown one of the Parsons turbines manufactured by Brown, Boveri & Company, Baden, Switzerland. The arrangement of the turbine blades and the balancing

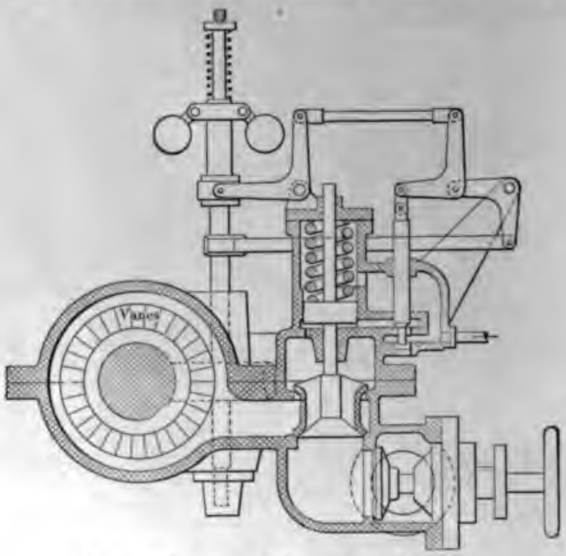


Fig. 13. Governor of Brown-Boveri Turbine.

pistons, including the increased diameter of the drum at the low-pressure end, are substantially the same as in the Parsons turbine as made by the Westinghouse Machine Company in this country. The governor, which is shown in Fig. 13, also operates on the same principle, following the lines laid down in the Parsons patent in 1896. The oiling is by forced lubrication and the general arrangement of the valves, governor, oil piping, etc., is clearly indicated in the engraving.

By-pass Valves.—Mr. Brown of this firm has taken out a United States patent for a by-pass valve to supply high-pressure steam to the low-pressure end of the turbine, which is somewhat

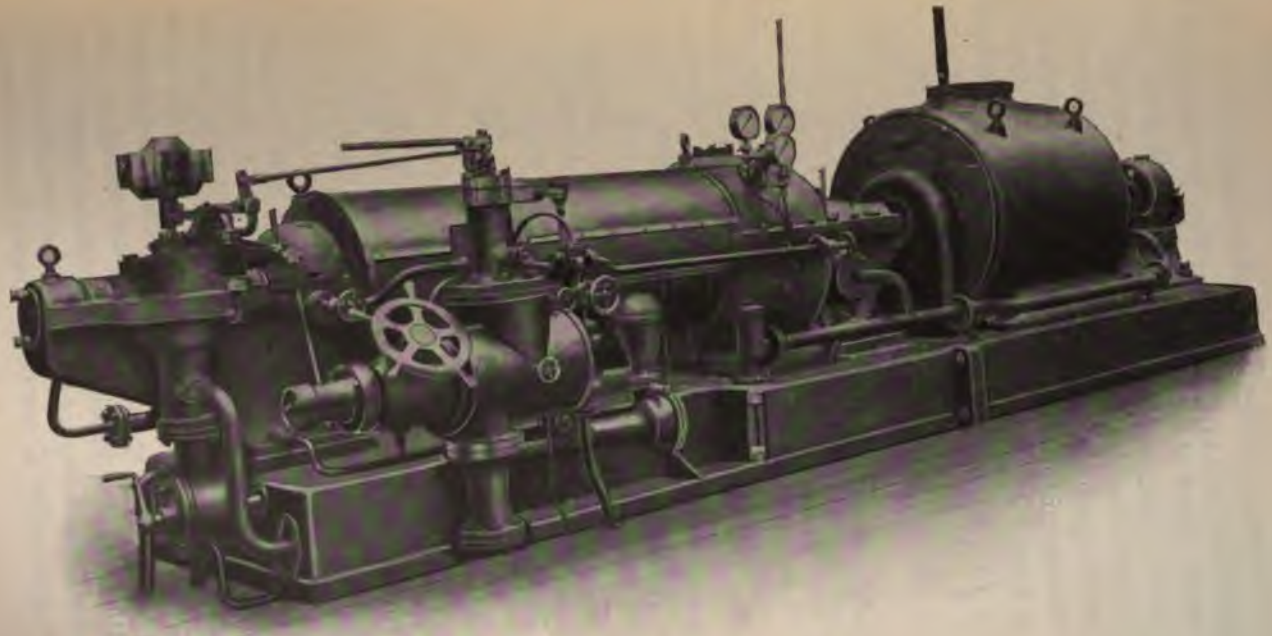


Fig. 14. Brown-Boveri Turbine.

different from the Hodgkinson valve already illustrated. The invention consists in providing pipes from the steam chest to the intermediate stages of the turbine and controlling the opening of these pipes by the governor.

In Fig. 15, steam is admitted through the supply pipe, *A*, and when the throttle valve, *B*, is raised by the governor, it will pass in the direction of the arrows through the valve, *C*, to the steam space, *D*, whence it will enter the high-pressure end of the turbine.

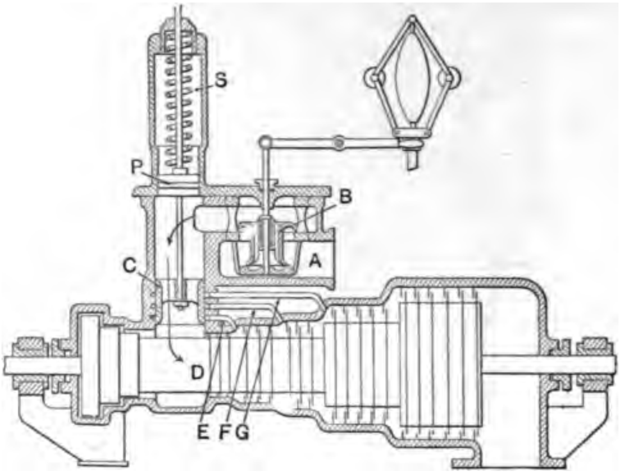


Fig. 15. By-Pass Arrangement.

The pipes, *E*, *F* and *G*, leading to other points in the turbine, are opened to the steam space by movement of the valve, *C*, which uncovers several ports shown in succession, when it is raised. Ordinarily, it is kept in its lowest position by the spring, *S*, thus closing all the ports. If, however, the speed of the turbine decreases under a heavy load the governor will admit a greater quantity of steam through valve *D* to the steam space, increasing the pressure in the steam space and forcing the piston, *P*, upward against the pressure of the spring, by this means raising the valve, *C*, and admitting steam successively to the different stages of the turbine. Brown's patent also covers an arrangement, for a similar purpose, having the valve, *C*, controlled directly by the governor.

The Allis-Chalmers Turbine.

The illustration, Fig. 16, shows the first turbine installed in this country of this type. It is now in operation at Utica, N. Y. It is rated at 1,500 Kw., normal load, and runs at a speed of 1,800 revolutions per minute. It is direct-coupled to an Allis-Chalmers two-phase, 60-cycle revolving-field alternator, operating at 2,500 volts. The unit has a continuous overload capacity of 25 per cent, with a 3-hour 50 per cent overload capacity without exceeding a safe generator temperature, and capable of a 100 per cent safe momentary overload. Artificial ventilation by means of an electrically driven fan blower will, however, enable the unit to be run safely beyond its rated overload capacity.

Blading.—The chief distinguishing feature is the blading. The roots of the blades are formed in dovetail shape by special machinery, and are inserted in slots cut in foundation or base rings; these slots being formed by special machine tools in such a way as to exactly conform to the shapes of the blade roots. The foundation rings themselves are of dovetail shape in cross section and are inserted in dovetailed grooves cut in the turbine casing and spindle respectively, in which they are held by key pieces, much in the same way that the well-known "Lewis bolt" is fastened. In order to further insure the integrity of the construction, the key pieces or rings after being driven into place are upset into undercut grooves.

Another noticeable feature of the blading is the method of reinforcing and protecting the tips of the blades, which is a point upon which much thought has been expended by various inventors. In forming the blades a shouldered projection is left at the tip. This is inserted in a slot punched in a shroud ring; the slots being punched by special machinery in such a way as to produce accurate spacing and at the same time form the slots so that they will give the proper angles to the blades independent of the slots in the base ring. After the blade tips are inserted in the slots in the shroud rings they are riveted over by specially arranged pneumatic machinery.

The shroud rings are channel shaped with outwardly projecting flanges which, after assembly in the turbine, are turned and bored to give the necessary working clearance. The flanges of the chan-

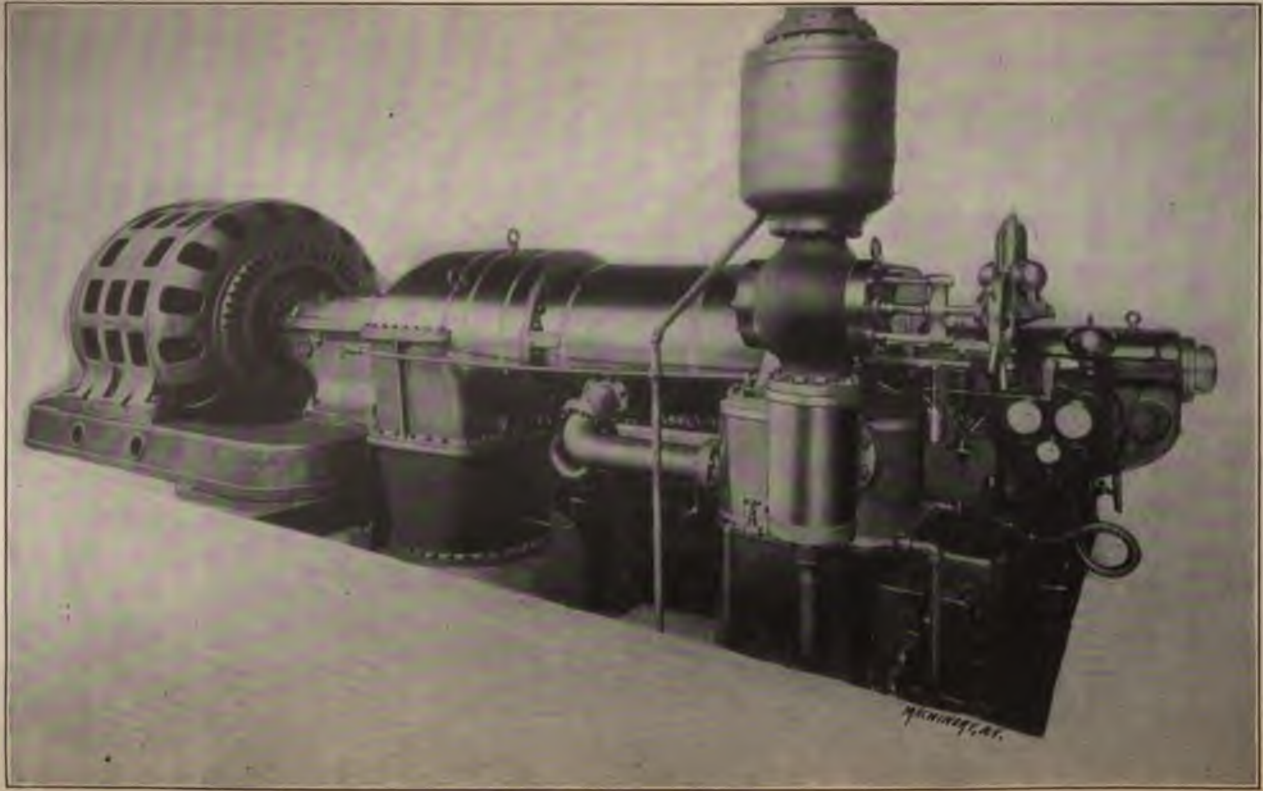


Fig. 16. Allis-Chalmers Turbine.

nels are made so thin that, although amply sufficient for stiffness, the shroud ring does not have the disadvantage of a solid shroud which acquires a dangerous temperature by friction in case of an accidental contact of the rotating and stationary parts. The use of a protecting shroud ring not only stiffens the blades, but enables the working clearance to be made smaller than in the case of naked blade tips, without danger in case of accidental contact, thus re-

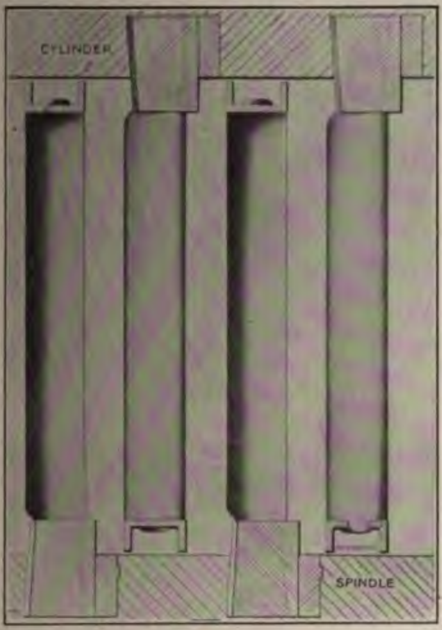


Fig. 17. Scheme of Blading.

ducing the leakage loss to a minimum. The shroud also acts as a safety device, protecting the blades in case of contact between stationary and rotating members, and preventing any individual blade from working loose and causing damage.

The entire blading is produced by machinery and is made up in half rings in the blading shop and carefully inspected before being inserted in the turbine. Fig. 17 shows the general scheme of the blading.

Balance Pistons.—Another special feature of this turbine is the

tion, a labyrinth packing of radial baffling type has been adopted, thus eliminating small axial clearance in this turbine. The advantage claimed for this construction is the use of smaller working clearances in the high-pressure and intermediate balance pistons.

Willans & Robinson Turbine.—The Allis-Chalmers Company have effected an alliance with the Turbine Advisory Syndicate of England, which includes Messrs. Willans & Robinson, the high-speed engine builders of Rugby, and several other well-known English firms. Their turbines are similar in construction to those manufactured by Willans & Robinson, both being made under the Fullagar patents. In Fig. 20 is a Willans & Robinson turbine with the top casing removed and which shows certain of the features of construction more clearly than the illustrations previously given. In this figure *A* is the rotor, *B* the top casing, and *C* the admission valve. *D* is the steam chest in which is the valve controlled by the governor, admitting steam through the pipe *E* to the high-pressure end and through pipe *F* to the low-pressure end in case of overloads. At *G* are two of the balance pistons, while at *H* is the small low-pressure balance piston previously referred to in the description of the Allis-Chalmers turbine. It will be noted that all connections, such as pipes and valves, are attached to the lower half of the casing, so that the upper half may be removed without disturbing any of the fittings.

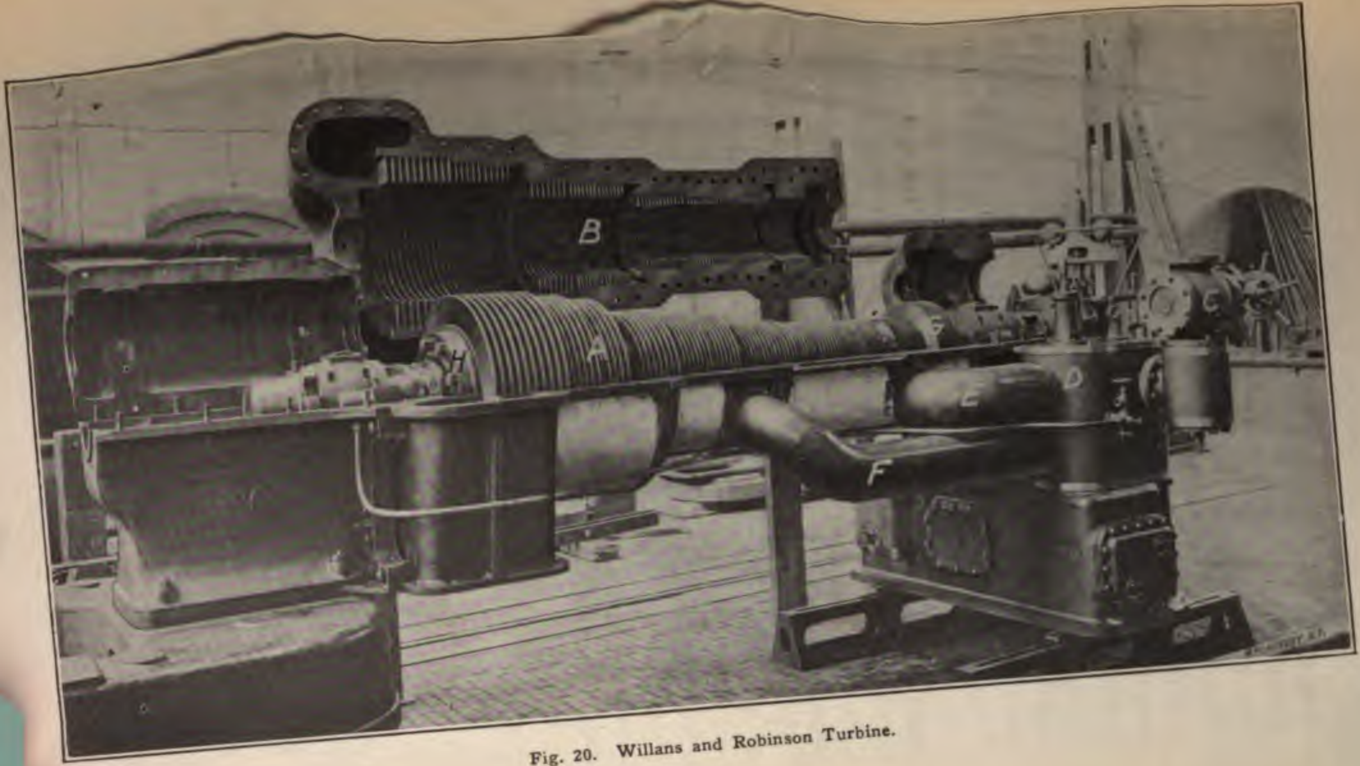


Fig. 20. Willans and Robinson Turbine.

has also been experimented with by The Westinghouse Machine Company in this country, the double-flow arrangement is adopted to eliminate end thrust and a long drum is obviated by having the turbine divided into two or more stages, in the first of which the steam acts on the impulse principle and in the second, or last, of which it acts on the reaction principle. Fig. 1, taken from the patent records, shows the general features of the design, although the turbine as actually constructed is different in some of the details.

Steam enters through the valve at the center and flows to the right and left through nozzles (not shown) in which it is expanded to a considerably lower pressure. The nozzles direct the steam against a series of impulse blades at *A*, consisting of two rows of moving and one row of stationary vanes. It then passes to the longer series of blades which may be entirely of the reaction or Parsons type; or may start at *B* with several rows of impulse vanes, divided into two or more pressure stages, and end with the reaction blading at *C*. A feature of the design is that the drum is of constant diameter, instead of in steps, as in the regular Parsons turbine, the reaction part of the drum corresponding to the last stage or step in the Parsons turbine and the expansion previous to this point being taken care of in the impulse section of the turbine.

As made by the British company, the drum or cylinder is a single forging carried on the spindle by a central supporting disk and stiffening disks of thin steel fitted into the ends. The impulse blades are carried by steel rings shrunk onto the drum and the reaction blades by grooves in the drum itself. Steam is expanded to about 60 pounds in the impulse section by means of the nozzles and two rows of rotating blades.

A vertical turbine, combining both the impulse and reaction principles, is manufactured by the Union Machine Company, Essen, Germany, and a horizontal one is manufactured by Sulzer Brothers, Winterthur, Switzerland. A sectional elevation of the latter is shown in Fig. 2. Steam enters through valve *A*, passing through the two impulse wheels at *B*. It then flows through the passages of the reaction wheel, *C*, thence around to the other side through the passages of reaction wheel *D*, to the exhaust space *E*.

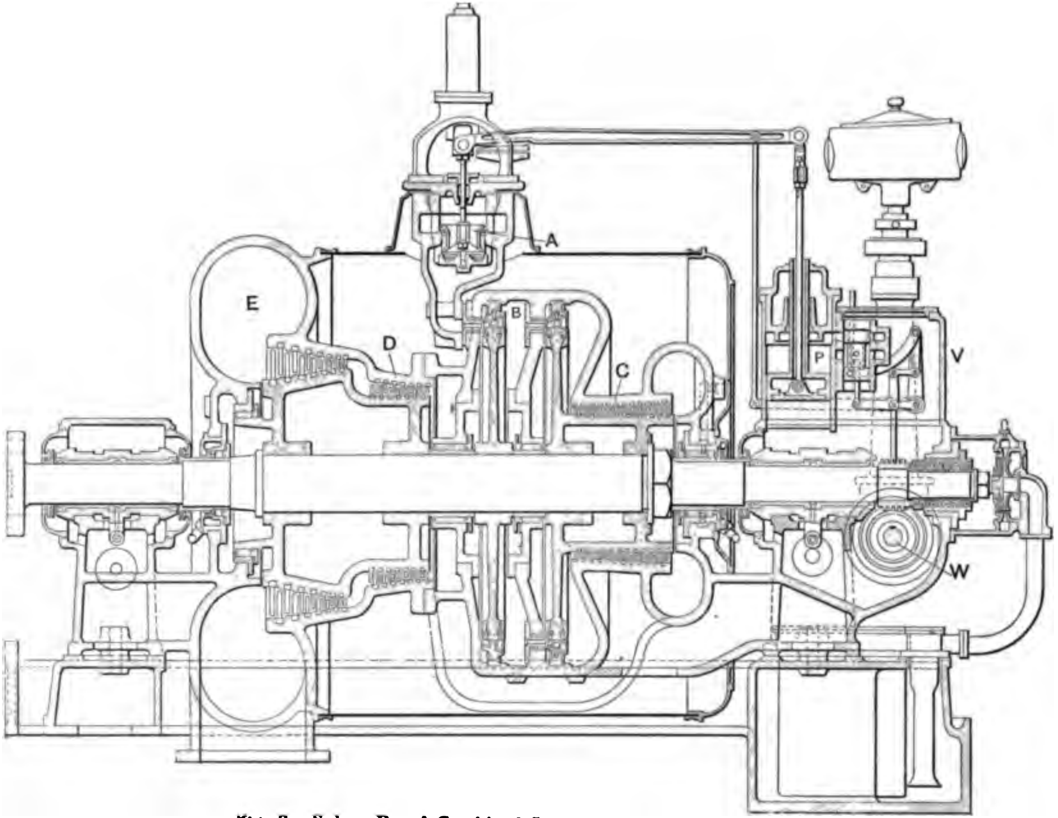


Fig. 2. Sulzer Bros.' Combined Impulse and Reaction Turbine.

It is designed that the thrust of reaction wheel *C* shall be balanced by that of wheel *D*. The admission valve *A* is an oscillating valve controlled by the piston *P*, which in turn is governed by the auxiliary valve *V*. This latter valve is given an oscillating motion by the wormwheel *W* and the position of the valve is at all times determined by the position of the governor.

The Lindmark Steam Turbine.

It has been known for some time that the De Laval Steam Turbine Company, Stockholm, Sweden, were developing a com-

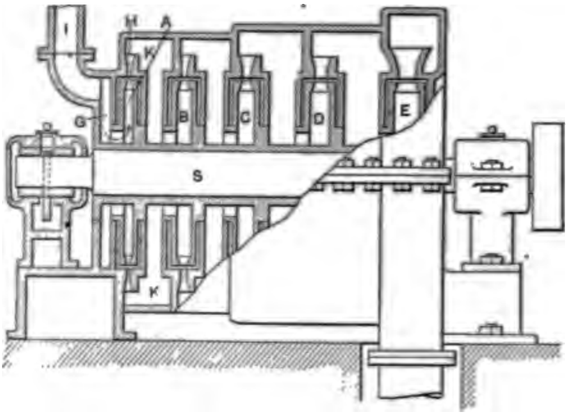


Fig. 3. Lindmark Turbine.

pound steam turbine for use where greater power is desired than it has been found advisable to attempt to supply with the single-wheel De Laval turbine. This new turbine is made under the patents of T. G. E. Lindmark, the first of which was issued in this country in 1902. Along substantially the same lines as the Lindmark inventions are patents taken out by P. J. Hedlund and consigned to the same company. At present no information concerning the Lindmark turbine is available other than given in the patent records, but inasmuch as a new principle is involved in the operation of this invention, it will be of interest to explain the features of this turbine in so far as is now possible.

Fig. 3 is a representation of a typical turbine containing the turbine wheels, *A*, *B*, *C*, *D* and *E*, attached to the shaft, *S*, and

rotating with it. Each wheel rotates in a separate compartment and is of the radial outflow type. The steam enters through an inlet pipe, *I*, to the steam space, *G*, whence it flows in the direction of the arrow into the first wheel, *A*. The two sides of the wheel converge at the periphery, forming a contracted outlet, and the steam discharges between curved vanes located in this contracted space, and flows through a diverging annular opening shown at *HH*, which guides the steam into a second compartment at *KK*. This process is repeated in connection with wheels, *B*, *C*, *D* and *E*.

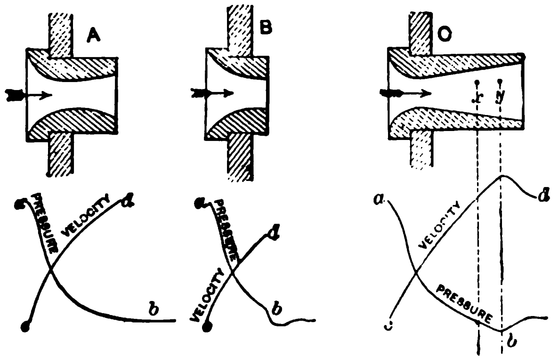


Fig. 4. Showing Principle of Nozzle Action.

Purpose of the Turbine.—The peculiarity of the turbine lies in the pressure and velocity effect upon the steam of the converging and diverging passages as the steam progresses through the turbine. The purpose of the inventor is to transform the potential energy and pressure of the steam into kinetic energy and velocity as it flows through the converging passages of any given wheel, and then to transform the residual kinetic energy and velocity of the steam after it discharges from the wheel into potential energy and pressure during the passage through the annular space, *HH*.

Flow of Steam Through Nozzles.—In order to explain the principle whereby this result is accomplished, it will be clearer to first refer to the action of steam in flowing through nozzles of different proportions. In Fig. 4 is shown a converging and diverging nozzle at *A*, through which steam flows from a higher to

a lower pressure in the direction of the arrow. The nozzle is supposed to be proportioned so as to give complete expansion, and the pressure will accordingly drop from *a* to *b*, while the velocity will increase from *c* to *d* as indicated by the diagram below the nozzle.

At *B* the diverging part of the nozzle is cut away so that the steam does not have the opportunity to fully expand until it has left the nozzle. The pressure, therefore, will gradually drop from *a* to *b*, at which latter point the steam leaves the nozzle and suddenly expands to the lower external pressure, producing a sudden change, *b*, in the curvature of the pressure line. Inasmuch as the expansion is not complete in the nozzle, the velocity line does not rise to as high a point as in the previous case.

At *C* the nozzle is shown lengthened out so as to produce over-expansion; that is to say, when the steam reaches point *x* in the nozzle it is expanded down to the external pressure of the medium into which the nozzle discharges, and beyond this point the steam will expand a few pounds below the outside pressure until, say, point *y* is reached. After this the pressure will rapidly rise again until it reaches the external pressure at the mouth of the nozzle.

The Action in the Lindmark Turbine.—It is this latter action which takes place in the diverging passages of the Lindmark turbine.

The application of the principle can best be explained by referring to the detail sectional sketch of the Lindmark turbine shown in Fig. 5, which represents half of one of the wheels and the connecting passages. Steam enters through the inlet and passes through the wheel in the direction of the arrow, and thence to the exhaust chamber, whence it escapes through the valve *B*. Suppose first the turbine wheel to be blocked so that it will not turn, valves *A* and *B* to be wide open, and the steam to flow freely through the turbine. At points 1, 2, 3, 4 and 5, are openings to which gauges may be attached for determining the pressure at the different points. The steam passages, *C* and *D*, will now act like a converging and diverging nozzle. At 1 the entering steam will be at boiler pressure; at 2 the pressure within the wheel casing will be substantially that at the throat of the nozzle, or about $\frac{6}{10}$ of the boiler pressure; at 3 it will be slightly lower; and

at 4 nearly as low as the pressure in the exhaust space indicated by the gauge at 5. (See "Steam Nozzles," Chap. I.)

Action, Wheel Blocked, Exhaust Valve Partially Closed.—Now, suppose valve *B* to be closed as much as possible without increasing the pressure, 2, within the wheel chamber. The pressure at 1 will obviously remain as before, and the expansion in the passages, *C*, of the wheel will, as before, carry the pressure at the throat of the nozzle to $^6/_{10}$ of the initial pressure. After the steam enters the diverging space, *D*, however, over-expansion will occur just as in nozzle *C*, Fig. 4. Probably the lowest pressure

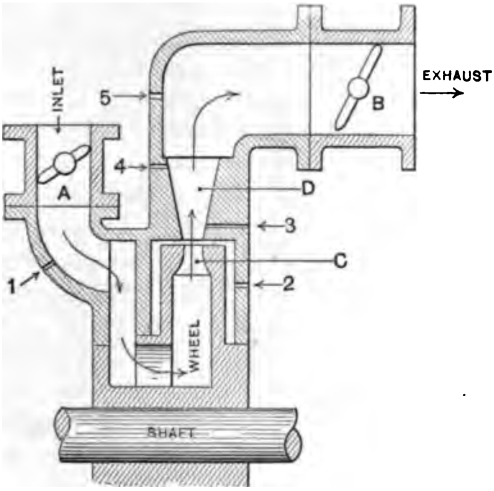


Fig. 5. Enlarged View of Passages.

will be at or near the point where the gauge is attached at 3, and from there on the velocity will decrease and the pressure will increase so that the gauges at 4 and 5 will indicate a pressure considerably higher than that in the wheel chamber at 2, but lower than the initial pressure at 1.

Action with the Wheel Rotating.—Finally if the wheel now be supposed to turn, instead of being blocked as before, and the exhaust valve still be partially closed, the principle of the steam's action will remain the same, the only change being due to the altered velocity of the steam owing to the fact that part of the velocity will be absorbed by the vanes of the rotating wheel.

Under these conditions the velocity of flow when the steam reaches the entrance to the diverging portion will be less than before and this will produce an effect, which, according to the experiments of Lindmark, will reduce the pressures somewhat at points 4 and 5.

The Lindmark turbine is a reaction turbine, since the expansion of the steam and increase in velocity occur in the passages of the wheel vanes and the curvature given to them is similar to that employed for reaction wheels of other types.

THE RATEAU STEAM ACCUMULATOR SYSTEM.

Turbines for Low-Pressure Steam.—One of the most interesting applications of the Rateau turbine is in employment with low-pressure steam supplied from engines working intermittently, such as rolling-mill engines, hoisting engines, etc. The majority of such engines are working under wasteful conditions.* They usually operate under widely varying loads, frequently with only a small degree of expansion and usually exhausting freely into the air. Professor Rateau has given special attention to the employment of waste steam from such engines and has obtained satisfactory results by means of his regenerative accumulator of steam combined with low-pressure turbines. The accumulator is intended to regulate the intermittent flow of steam before it passes to the turbine, and consists essentially of a tank containing solid or fluid materials which play the part of a flywheel for heat.

Description of the Accumulator.—In his paper before the American Society of Mechanical Engineers in June, 1904, Professor Rateau gave the following description of his accumulator:

The steam collects and is condensed as it arrives in large quantities in the apparatus, and is again vaporized during the time when the exhaust of the principal engine diminishes or ceases. The necessary variations for con-

*Trials of modern winding engines frequently show a steam consumption of 65 to 80 pounds per horse-power in mineral hoisted, while 100 to 120 pounds is not uncommon, so that allowing 20 per cent condensation in the cylinders and passages of the engine, there is discharged into the atmosphere by such an engine a minimum of 9,000 to 10,000 pounds of steam per hour which is totally lost. This steam is theoretically capable of developing 500 to 600 horse-power if supplied at atmospheric pressure and exhausted at a vacuum of 27 inches. Rolling mill engines frequently consume 50,000 pounds of steam per hour, which is theoretically capable of developing over 2,500 horse-power in expanding from atmospheric pressure to the vacuum of an ordinary condenser, even after deducting 20 per cent for condensation losses.—From paper by Leonce Battu, read at a meeting of the Western Society of Engineers, September, 1904.

densation and regeneration of the steam correspond to fluctuations in pressure in the accumulator, this pressure rising when the apparatus is being filled and descending when it is discharging into the turbine. Water which has a very high calorific capacity has been used as a heat fly-wheel, but in order to rapidly communicate to a liquid mass a considerable quantity of

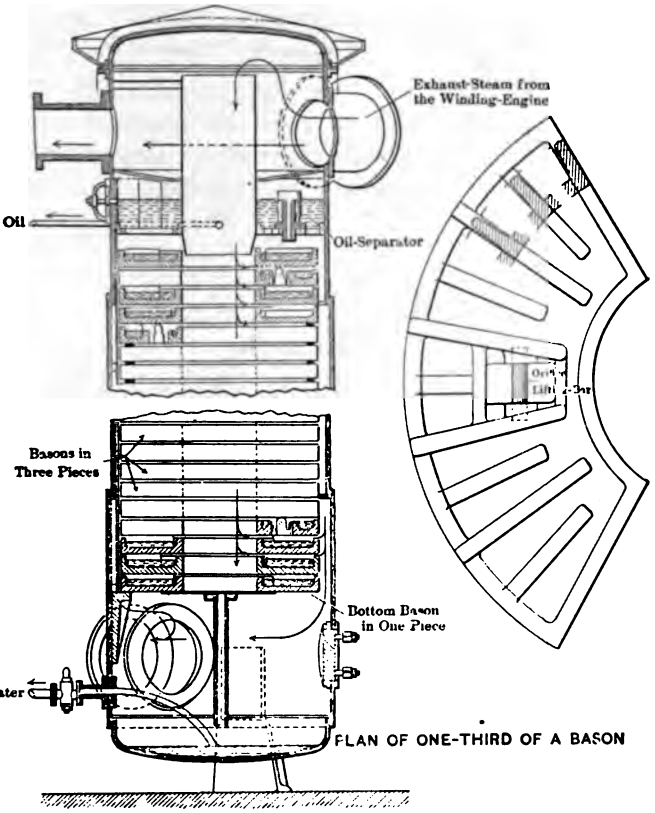


Fig. 6. Rateau Steam Accumulator with Iron Trays.

heat corresponding to the latent heat of steam to be condensed it becomes necessary, owing to the poor conductivity of water, either to arrange it in thin layers or to cause a rapid circulation in order to increase the surface of contact between the steam and the water itself. The first solution of the problem gave rise to the accumulator with flat cast-iron trays in which water is contained in shallow vessels arranged one above the other, Fig. 6. The second solution of the problem gave rise to the accumulator with water only in which a rapid circulation was produced by the injection of steam

into the body of the liquid itself, Fig. 7. The low-pressure turbine, fed by the regular flow which comes from the accumulator, and working, for example, between an admission pressure of 15 pounds per square inch and a vacuum at the condenser of 27 inches of mercury (back pressure of 1.6 pounds) can furnish an electric horse-power for about 31 pounds of steam per hour. In steel works, where reversible steam rolls are employed consuming about 45,000 pounds of steam per hour, it will be easy to develop, by means of accumulators and turbines, an extra output of over 1,100 electric horse-power.

Accumulators are fitted with several accessories which are necessary for their successful operation. One of these is an

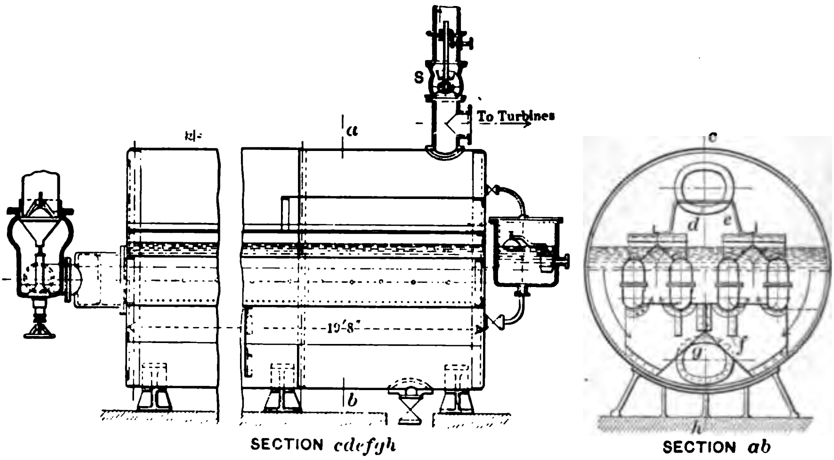


Fig. 7. Accumulator with Water only.

automatic relief valve to allow the steam from the engine to escape into the atmosphere of the condenser if the turbine should not require all of the steam exhausted by the main engine. Another is an automatic expansion valve provided so that live steam from the boiler may be admitted to the turbine, should the main engine be temporarily shut down or if sufficient exhaust steam is not available. There should also be a steam check valve and a water check valve, the former for shutting off the accumulator from the turbine when the main engine is shut down and the turbine is supplied with live steam only; while the latter is used to prevent the water in the accumulator returning toward the main

engine, through the exhaust supply pipe, when the engine is shut down.

*Calculations for an Accumulator.**—To calculate the weight of water or cast iron necessary for an accumulator, we must know: (1) The total weight of steam required by the turbine per hour; (2) assume a length of time for the accumulator to operate without a steam supply; and (3) assume an allowable drop in temperature of the accumulator while the steam supply is shut off and the turbine is in operation. For example, suppose the turbine to consume 2,200 pounds of steam per hour; the duration of the stop to be one minute; and the range of temperature 10 degrees F.

Let G=weight of steam used by turbine during stoppage of steam supply.

$$\text{Then } G=\frac{2{,}200\times 1}{60}=36.6.$$

At atmospheric pressure one pound of steam contains 966 B. T. U. Whence, the accumulator must be able to deliver

$$36.6\times 966=35{,}400 \text{ B. T. U.}$$

Since the specific heat of water is one, the weight of water necessary to deliver this quantity of heat with a range of temperature of 10 degrees F. is

$$\frac{35{,}400}{10}=3{,}540 \text{ pounds, or about 1.5 tons.}$$

If the accumulator were to contain cast iron instead of water, the specific heat of which is 0.11, a weight of iron would be required about 9 times as great as for water. Hence, for cast iron we have

$$\frac{1.5}{0.11}=13.75 \text{ tons.}$$

Tests on Rateau's Accumulator System.—The first plant of this type to be installed was at the Bruay Mines, France. The accumulator was of the type shown in Fig. 6, and a drawing of the

*Paper by Leonce Battu, Western Society of Engineers, September, 1904.

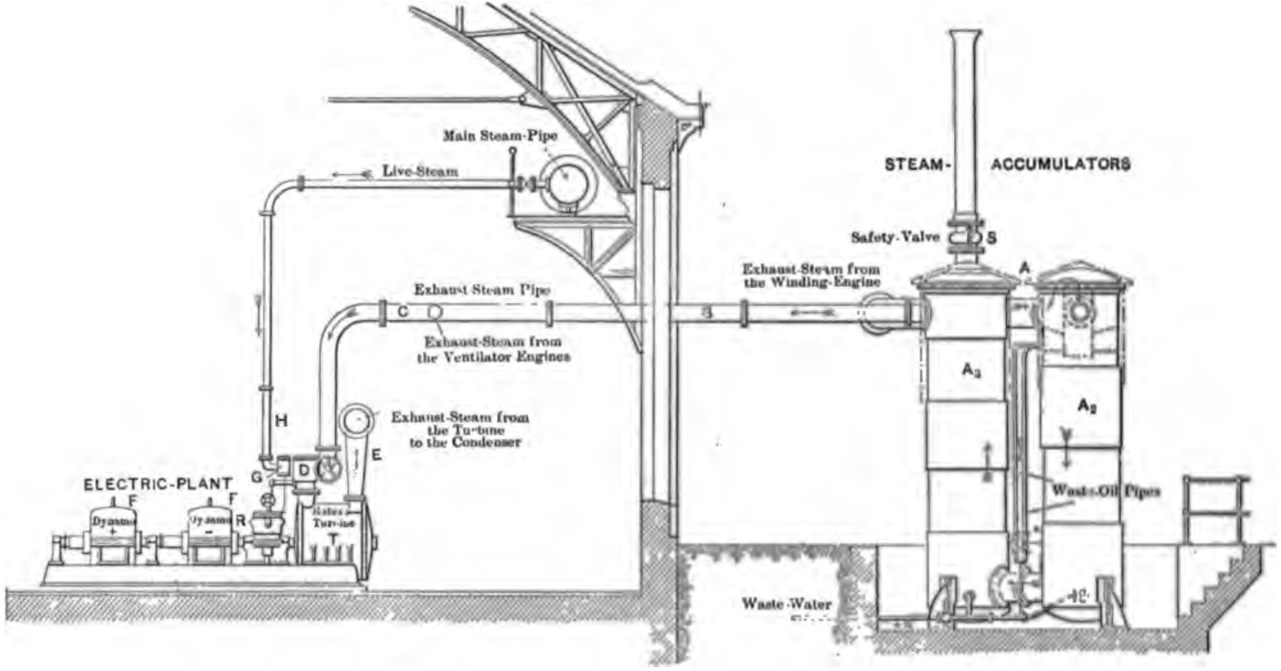


Fig. 8. Turbine Plant with Accumulators.

complete plant is shown in Fig. 8. This plant is of 3,000 horse-power, and tests taken at the time the machinery was installed and 15 months afterwards showed no diminution of efficiency. In one series of tests the pressures of the steam supplied to the turbine ranged from 12 to 14.5 pounds absolute, and the condenser pressures ranged from 2.13 to 2.62 pounds absolute. The consumption of steam per horse-power hour, the horse-power being the electric horse-power delivered by the generator, ranged from 37.4 to 45.2.

In his paper previously referred to M. Battu gives the following estimate of the probable consumption of steam operating on this system:

The table shows the steam consumption required to give a horse-power electric at the terminals of the generators:

Absolute pressure of steam, at admission..........	30 lbs.	15 lbs.	7.5 lbs.
Vacuum in the Condenser of 27.6 inches.........1.14 lbs.	21.7	26.4	35.3
" " " " " 25.0 " ....1.85 "	23.6	31.7	47.3
" " " " " 21.6 " ....2.56 "	26.4	36.3	61.6

The first horizontal line of the table corresponds to the case where a surface condenser giving a vacuum of 27.6 inches is used. The second to an ordinary jet condenser giving a vacuum of 26 inches, and the third to an ejector condenser giving a vacuum of 24.6 inches. These vacuums are easily obtained in practice.

The central vertical column corresponds to the usual conditions when the steam is supplied to the turbine at atmospheric pressure. The first column gives the consumption in the case where the primary engines exhaust against a gauge pressure of fifteen pounds, in order to take advantage of the superior efficiency with which it can be used in the turbine, while the third column represents the condition where it is desirable not to disturb the condensing operation of the main engines, and consequently to utilize steam exhausting from them at about 7½ pounds absolute.

Low-Pressure Curtis Turbine.

The Philadelphia Rapid Transit Company have installed an 800 Kw. Curtis low-pressure turbine in their station at Thirteenth and Mt. Vernon Streets. This station is equipped with Corliss engines, and as it is located midway between the Schuylkill and Delaware rivers the engines have always been run non-condensing. An Alberger condenser and cooling tower have been installed,

however, for use with the turbine and a vacuum of 28 inches or more has been maintained during the cool weather. The turbine has four wheels, each with a single row of buckets. When the turbine is receiving steam at atmospheric pressure, without moisture, the guarantees provide that the steam consumption shall not exceed 36 pounds per kilowatt hour at full load and 40 pounds at half load, back pressure being two inches. At four inches back pressure these figures are respectively 45 and 50 pounds. Tests made on the machine at the factory showed even better performance. It is estimated that the turbine will increase the output of the engine or engines that supply steam to it about 66⅔ per cent instead of the 25 per cent usually expected from the application of a condenser.

CHAPTER IX

STEAM TURBINE PERFORMANCE—COMPARISONS WITH THE STEAM ENGINE.

There will be found in this chapter the results of a number of tests upon turbines of different types. In these the steam consumption is usually given in pounds per electrical horse-power per hour, or in pounds per kilowatt hour. In the majority of cases turbines are direct-connected to electric generators, and the power is most readily measured by the electrical instruments of the switchboard, which show the output of the generator instead of the actual power developed by the turbine. In factory tests, however, before the turbine is shipped, the brake horse-power is often determined, since means are usually at hand for attaching an absorption dynamometer. This plan is followed at the works of the Westinghouse Machine Company, Pittsburg, Pa.

Kilowatts and Electrical Horse-Power.—The power delivered by the generator is expressed in kilowatts or in electrical horse-power, the latter being the equivalent, in electric units, of mechanical horse-power. For converting kilowatts to horse-power and horse-power to kilowatts, we have:—

1 kilowatt=1.3405 horse-power=1.34 horse-power, nearly.

1 horse-power=0.7459 kilowatt=0.746 kilowatt, nearly.

Table I. will be of assistance in converting the more usual values of kilowatts to electrical horse-power and of electrical horse-power to kilowatts.

Table II. gives steam consumption in pounds per electrical horse-power hour corresponding to steam consumption per kilowatt hour, taken at half-pound intervals, within the limits usually met with in practice.

Table III. gives steam consumption per kilowatt hour corresponding to consumption per electrical horse-power hour.

Indicated, or Internal Horse-Power.—There is no such thing as the indicated horse-power of a turbine, because no indicator has been, and probably none can be, devised to show the internal power developed. While an indicator might show the energy of a

jet of steam discharged from a nozzle, it would be practically impossible to register the amount of energy given up by a jet to the blades of a compound turbine, where the losses might be greater or less, according to the design, load, and other running conditions.

Engineers are so familiar with the water rates of reciprocating engines on the basis of the indicated horse-power, that in comparing a turbine with an engine it is usual to reduce the figures for the steam consumption of the turbine to terms of the indicated horse-power of a reciprocating engine having the same electrical output, or brake horse-power, as the case may be.

TABLE I.

CONVERSION OF HORSE POWER INTO KILOWATTS AND KILOWATTS INTO HORSE POWER.

1 Kw. = 1.3405 H. P. (1.34).

1 H. P. 0.7459 Kw. (0.746).

Number.	Kilowatts to Horse Power.	Horse Power to Kilowatts.	Number.	Kilowatts to Horse Power.	Horse Power to Kilowatts.
1	1.34	0.75	36	48.26	26.85
2	2.68	1.49	37	49.60	27.60
3	4.02	2.24	38	50.94	28.34
4	5.36	2.98	39	52.28	29.09
5	6.70	3.73	40	53.62	29.84
6	8.04	4.48	41	54.96	30.58
7	9.38	5.22	42	56.30	31.33
8	10.72	5.97	43	57.64	32.07
9	12.06	6.71	44	58.98	32.82
10	13.40	7.46	45	60.32	33.57
11	14.75	8.20	46	61.66	34.31
12	16.09	8.95	47	63.00	35.06
13	17.43	9.70	48	64.34	35.80
14	18.77	10.44	49	65.68	36.55
15	20.11	11.19	50	67.03	37.30
16	21.45	11.93	75	100.5	55.94
17	22.79	12.68	100	134.1	74.59
18	24.13	13.43	150	201.2	111.9
19	25.47	14.17	200	268.1	149.2
20	26.81	14.92	250	335.1	186.5
21	28.15	15.66	300	402.2	223.8
22	29.49	16.41	350	469.2	261.1
23	30.83	17.16	400	536.2	298.4
24	32.17	17.90	450	603.2	335.7
25	33.51	18.65	500	670.3	373.0
26	34.85	19.39	750	1005.	559.4
27	36.19	20.14	1000	1341.	745.9
28	37.53	20.88	1250	1675.	932.4
29	38.87	21.63	1500	2011.	1119.
30	40.22	22.38	2000	2681.	1492.
31	41.56	23.12	2500	3351.	1865.
32	42.90	23.87	3000	4022.	2238.
33	44.24	24.61	4000	5362.	2984.
34	45.58	25.36	5000	6703.	3730.
35	46.92	26.11			

TABLE II.

RELATIVE STEAM CONSUMPTION PER KILOWATT HOUR AND ELECTRICAL HORSE POWER HOUR.

Lb. per Kw. Hour.	Corresponding lb. per E. H. P. Hour.
13	9.70
13.5	10.07
14	10.44
14.5	10.82
15	11.19
15.5	11.56
16	11.93
16.5	12.33
17	12.68
17.5	13.06
18	13.43
18.5	13.80
19	14.17
19.5	14.55
20	14.92
20.5	15.29
21	15.66
21.5	16.04
22	16.41
22.5	16.79
23	17.16
23.5	17.53
24	17.90
24.5	18.28
25	18.65
25.5	19.02
26	19.39
26.5	19.77
27	20.14
27.5	20.51
28	20.88
28.5	21.26
29	21.63
29.5	22.01
30	22.38
30.5	22.75
31	23.12
31.5	23.50
32	23.87
32.5	24.24
33	24.61
33.5	24.99
34	25.36
34.5	25.74
35	26.11

TABLE III.

RELATIVE STEAM CONSUMPTION PER ELECTRICAL HORSE POWER HOUR AND KILOWATT HOUR.

Lb. per E. H. P. Hour.	Corresponding lb. per Kw. Hour.
10	13.40
10.5	14.08
11	14.75
11.5	15.42
12	16.09
12.5	16.76
13	17.43
13.5	18.10
14	18.77
14.5	19.44
15	20.11
15.5	20.78
16	21.45
16.5	22.12
17	22.79
17.5	23.46
18	24.13
18.5	24.80
19	25.47
19.5	26.14
20	26.81
20.5	27.48
21	28.15
21.5	28.82
22	29.49
22.5	30.16
23	30.83
23.5	30.16
24	32.17
24.5	32.84
25	33.51
25.5	34.18
26	34.85
26.5	35.52
27	36.19

*Comparing Turbine Performance with Engine Performance.** —Calculations of this character must take into account the efficiencies of engines and generators, data upon which will shortly be given. The actual calculations involve nothing more difficult than the principles of percentage.

Example: Let a turbine unit deliver 500 electrical horse-power, and consume 7,500 pounds of steam per hour. Its rate of steam consumption will then be 7,500÷500=15 pounds per electrical horse-power per hour. What would be the indicated horse-power, and the consumption per indicated horse-power per hour, of a reciprocating engine having the same rate of consumption per electrical horse-power per hour? Assume the engine to be direct-connected to a generator, the efficiency of the generator to be 95 per cent, and the mechanical efficiency of the engine 94 per cent. The combined efficiency will then be 0.94×0.95=0.89. The indicated horse-power=500÷0.89=561.8. The steam consumption per indicated horse-power per hour=7,500÷561.8=13.4 pounds. The latter could have been obtained directly by multiplying the water rate for the turbine, 15 pounds per electrical horse-power per hour, by 0.89, thus: 15×0.89=13.4.

It is to be noted that in these comparative calculations, where we estimate engine performance for comparison with turbine results, we use the efficiency of *the engine-driven generators,* not of turbine generators.

*In a paper upon the Curtis turbine, read by Chas. B. Burleigh before the New England Railroad Club, April, 1905, are calculations of the losses in engines and generators, as follows: Let us take a Curtis turbine guarantee of 20 pounds of steam per kilowatt hour and figure what the reciprocating engine guarantee per indicated horse-power should be to just equal it. Twenty pounds per kilowatt hour is equivalent to 20×.746=14.92 pounds per electrical horse-power hour. To start with, we must make our turbine test with instruments mounted on the switchboard, and the loss in the conductors from the generator to the switchboard being 1 per cent, we must deduct this 1 per cent of 14.92, or 0.1492 pounds. 14.92—0.1492= 14.77 pounds. Next we must deduct the generator loss, which, since the generator is designed to meet the engine speed, is in most cases more than as though the ideal generator could have been used and the generator adapted to it. We should, therefore, allow at least 5 per cent generator loss. Thus, 5 per cent of 14.77=0.74 and 14.77—0.74=14.03 pounds per brake horse-power. We are now back to the engine, but the indicator card does not take into account the friction losses in the engine, so these must be deducted, and I think you will agree with me that 7 per cent is fair for this. Seven per cent of 14.03=0.98 and 14.03—0.98=13.05 pounds per indicated horse-power. Therefore:

A turbine guarantee of 20 pounds of steam per kilowatt hour,
An engine guarantee of 13.05 pounds of steam per indicated horse-power hour,
Or a turbine guarantee of 14.92 pounds of steam per electrical horse-power hour,
are identical.

Efficiencies of Engine-type Generators.—Table IV. has been prepared from data furnished by manufacturers of generators.

TABLE IV.

EFFICIENCIES OF ALTERNATING AND DIRECT CURRENT ENGINE-TYPE GENERATORS.

ALTERNATING CURRENT GENERATORS (ABOUT 2300 VOLTS).

Kw.	Speed.	Per Cent Efficiency.				
		¼ Load.	½ Load.	¾ Load.	Full Load.	1¼ Load.
500	120	86	91	93	94	95
1500	100	87	92	94	95	96
3000	75	88	93	95	96	96.5
5000	75	89	94	95.5	96.5	96.7

DIRECT CURRENT GENERATORS.

Kw.	Volts.	Per Cent Efficiency.			
		¼ Load.	½ Load.	¾ Load.	Full Load.
100	125	86	91	91.5	92
500	250	88	92	93.5	94
1000	250	89	92.5	93.5	94
1500	650	91	94	94.5	95
2700	575	91	94	94.5	95

Efficiencies of Turbine Generators.—In estimating the brake horse-power of a turbine, having given the electrical horse-power, or vice versa, there must be an allowance for the efficiency of the generator driven by the turbine.

In the De Laval turbine outfits, twin generators are used, which reduces the size of each generator by about one half, and the efficiencies are low on this account. Medium size generators for these turbines, say of 200 Kw. capacity, have an efficiency ranging from 88 to 91 per cent between one half and full load, if for direct current. In alternating-current units of the same capacity, the efficiency varies from 86 to 92 per cent, between one half and full load.

A test upon a 1,250 Kw. A. C. generator for use with a Westinghouse-Parsons turbine, reported by A. W. Mattice in the

Electrical World, February 20, 1904, showed efficiencies of 86 per cent at quarter load, 93 per cent at half load, and 96 per cent at full load. Also, a report of a test upon a 400 Kw. A. C. generator for Parsons turbine, normal voltage of 440, by F. P. Sheldon & Co., Providence, R. I., contains the following figures:

	Guaranteed Efficiency.	*Measured Efficiency.*
Full load	94.5 per cent	96.6 per cent
¾ "	93.5 "	95.7 "
½ "	91 "	94.6 "

Tests on an Allis-Chalmers 5,500 Kw. A. C. turbo-generator, reported in *The Engineer*, February 1, 1906, showed the following efficiencies:

½ load,	97 per cent	full load,	98.3 per cent
¾ "	97.9 "	1¼ "	98.5 "

Alternating current generators of the type used with the Curtis turbine, 500 Kw. and over, have efficiencies ranging from 96 to 97.5 per cent at full load. These generators have a high electrical efficiency, because the Curtis turbine runs at a favorable speed for the generator; and a high mechanical efficiency, owing to the small friction of the vertical shaft.

Mechanical Efficiency of Steam Engines.—The importance of being able to make a just comparison between the performance of steam turbines and steam engines makes it essential to carefully consider the subject of engine friction, so that proper allowances may be used when reducing electrical horse-power to equivalent "internal" horse-power.

The friction loss of steam engines remains very nearly constant at all ordinary operating loads. Professor Thurston has treated the subject exhaustively in papers to be found in the *Transactions of the American Society of Mechanical Engineers*, volumes VIII., IX., and X., and his conclusions, as well as those of others who commented on his investigations, were that it is substantially correct to consider the friction loss constant under varying loads. He found this loss to be influenced to a much greater extent by the degree of lubrication, change in speed, steam

distribution, and design and condition of the engine, than by a change in load.*

If the friction load is constant, the efficiency of an engine will evidently vary widely with change in load. Thus, if a constant friction load is 10 per cent at full load, it will be 20 per cent at half load and the efficiencies will be 90 and 80 per cent.

A few tests will now be quoted to indicate what allowances may be made for the mechanical efficiency of engines of different types.

Friction Tests of Steam Engines.

1. A 5,400 horse-power, three-cylinder, vertical, compound Westinghouse engine** at the Waterside station of the New York Edison Company had a combined efficiency of engine and generator of 94.5 to 95.2 per cent under such variation in load as the engine was called upon to carry. The friction load was 118.6 horse-power, or 2.2 per cent of the normal power of the engine.

2. A 3,500 horse-power, triple-expansion, horizontal engine† at the Berlin electricity works had a mechanical efficiency of from 89.5 to 92.9 per cent under running loads. When running empty the friction horse-power was 266.3, or 7.6 per cent of the nominal power of the engine.

3. A 2,500 horse-power, cross-compound, horizontal Allis engine‡ at the Harvard Square power station, Cambridge, Mass., had a combined efficiency of engine and generator (direct current) of 90 per cent at an electrical load of 2,200 horse-power.

4. An 850 horse-power, cross-compound, horizontal Rice and Sargent engine§ (Corliss type) gave the following results: At normal load the combined efficiency of generator and engine was 93 per cent; at 1,000 horse-power, 94 per cent; at 627 and 490 horse-power, 90 per cent; at 340 horse-power, 83 per cent. The differences between indicated and electrical horse-power at the above loads, taken in their order, were: 57, 59, 58, 63, 51, and 57. The friction load, dynamo running idle, was 45 horse-power, or 5.3 per cent of the rated power of the engine.

*See friction tests "No. 4," which follow in the main text. In these tests the frictional and electrical losses between the cylinder and switchboard, equal to the differences between the indicated and electrical power at different loads, vary from 51 to 63 horse-power, in amounts which bear no relation to the power developed. This is a common experience in all friction tests of engines. It is impossible to trace any relation between the load and the friction loss. More tests can probably be quoted to show that the friction loss is constant, or nearly so, than to support any other supposition.

***Engineering Record*, May 28, 1904.
Since the above was in type a test has been reported upon one of the 7,500 horse-power, twin, vertical-horizontal, Reynolds-Corliss engines of the Interborough Rapid Transit Company, New York City. The apparent combined efficiency of engine and generator was 92.4 per cent.

†*Traction and Transmission*, London, January, 1902.

‡*Technology Quarterly*, September, 1898.

§Test by Prof. D. S. Jacobus.

5. In Prof. Thurston's papers upon the friction loss in steam engines previously referred to in this chapter, are reports of several tests: (a) A 50 horse-power, Straight Line, high-speed engine, non-condensing, had a constant friction load of about 3 horse-power, or 6 per cent of *full* load, giving an efficiency of 94 per cent. (b) A compound, condensing engine had a friction horse-power of 44 when developing 347 horse-power and a friction horse-power of 40 when developing 185 horse-power. (c) A 16 x 30 Porter-Allen engine had a friction horse-power of 12.7 when developing 142 horse-power and 8.4 at 84 horse-power. The last two engines (b and c) probably had full load efficiencies of about 87 and 90 per cent, respectively.

6. In addition to the above, reference may be made to the internal friction of large pumping engines, which is usually about 10 per cent. The famous Leavitt pumping engine at the Boston sewage works showed an efficiency of 90 per cent, and the Chestnut Hill, Mass., engine by the same designer had an efficiency of 93 per cent.

Summary of Engine Friction Tests.—Tests 1, 3 and 4 give the combined efficiency of engine and generator, for three large units, ranging from 850 to 5,400 horse-power, one of which is vertical. These are all modern, Corliss-type engines. Taking the two horizontal engines, the efficiency of No. 3 is 90 per cent and of No. 4 93 per cent, at about normal load. The average efficiency of No. 4 at the different loads is 90 per cent. It would seem that for large engines of this type an estimate of 90 per cent would be conservative for combined efficiency at normal load. For vertical engines the combined efficiency would be higher, reaching 94 per cent (as a safe figure) in the large sizes.

In the light of the above efficiency tests the following table of the mechanical efficiency of engines has been prepared:

TABLE V.

MECHANICAL EFFICIENCY OF ENGINES AT OR NEAR THEIR NORMAL LOAD.

Engine.	Efficiency, Per Cent. Engine Alone.	Efficiency, Per Cent. Engine and Generator.
Large Vertical Corliss, Compound	93.5 to 97.5	92 to 94
Large Horizontal Corliss, Compound	94.5	90
Large Horizontal, Triple	91	
High Speed, Simple	94	
" and Medium-sized Horizontal, Compound	87 to 93	
Pumping	90	

The Thermal Unit Basis of Performance.—In comparing the results of engine and turbine tests, and especially where boiler pressures differ or superheated steam is used, the efficiencies should be calculated on the basis of the heat units contained in the steam. Under conditions of varying pressure or of superheat the pounds of steam per horse-power per hour do not indicate the amount of heat energy contained in the steam.

Calculations are given herewith to illustrate the heat-unit method, taken from a report of tests upon a 400 Kw. Westinghouse-Parsons steam turbine, by Messrs. Dean and Main. The efficiencies for superheated steam are figured by using 0.48 as the value for the specific heat of superheated steam. These calculations will prove of assistance should the reader desire to recalculate any tests on the heat-unit basis.

CALCULATION OF EFFICIENCIES OF 400 KW. TURBINE.

	Dry Steam.	100 Degrees Superheat.	180 Degrees Superheat.
Brake horse-power developed	593.17	594.60	592.27
Corresponding indicated or internal horse-power of a reciprocating engine = B. H. P. ÷ 0.94	631.03	632.55	630.07
Total steam used per hour, pounds	8,249	7,384	6,779
Steam used per internal horse-power per hour, pounds	13.07	11.67	10.76
Absolute steam pressure, pounds	168.57	169.98	167.84
Superheat (exact figures)	0	109 Deg. F.	181 Deg. F.
Temperature condensed steam	95.3 Deg. F	95.8 Deg. F.	94.7 Deg. F.
Heat in one pound of dry saturated steam at above pressures, B. T. U.	1194.1	1194.3	1193.9
Heat in superheat per pound, B. T. U. (on the basis of specific heat = 0.48)	0	52.3	86.9
Total heat in one pound of steam, B. T. U.	1194.1	1246.6	1280.8
Heat of liquid in condensed steam, B. T. U.	63.3	63.8	62.7
Heat used by turbine per pound, B. T. U.	1130.8	1182.8	1218.1

From the above the following results are obtained:

Case of Dry Steam.

B. T. U. used by turbine per minute $= (1{,}130.8 \times 8{,}249) \div 60 =$ 155,466 B. T. U.

B. T. U. used per internal horse-power per minute, 155,466÷631.03=246.37 B. T. U.

$$\text{Thermal efficiency}=\frac{33,000}{246.37\times778}=17.22 \text{ per cent.}$$

Case of 100° Superheat.

B. T. U. per minute, (1,152×7,584)÷60=145,563 B. T. U.

B. T. U. used per internal horse-power per minute,

145,563÷632.55=230.12 B. T. U.

$$\text{Thermal efficiency},\ \frac{33,000}{230.12\times778}=0.1843=18.43 \text{ per cent.}$$

Case of 150° Superheat.

B. T. U. per minute, (1,218.1×6,779)÷60=137,625 B. T. U.

B. T. U. used per internal horse-power per minute,

137,625÷639.07=215.33 B. T. U.

$$\text{Thermal efficiency},\ \frac{33,000}{215.33\times778}=0.1943=19.43 \text{ per cent.}$$

Results of Turbine Tests.

Tables VII. to XVII., inclusive, contain results of tests upon turbines of different types and show in condensed form what economy may be expected of turbines operating under different conditions.

TABLE VI.

TESTS ON 30 H. P. DE LAVAL TURBINE.*

Non-Condensing. Initial Pressure 99.54 lb. Absolute.

	Half Load.		Full Load.	
	Saturated Steam.	Super-heated Steam.	Saturated Steam.	Super-heated Steam.
Temperature of Steam Degrees F	327	800	327	932
Brake horse power	21.1	24.2	43.5	51.2
Lb. steam per B. H. P. per hour	43.3	31.5	39.6	25.7
Temperature Exhaust Degrees F	2·2	588	212	649

*Made at Polytechnical College, Dresden.

TABLE VII.

TESTS ON DE LAVAL TURBINES AT DIFFERENT LOADS.*

Turbine Machine.	Initial Pressure. Lb. per sq. inch.	Vacuum, inches.	No. of nozzles open.	Electrical H. P.	Lb. steam per electrical H. P. per hour.	Remarks.
100 H. P. Turbine Dynamo.	103.7	25.8	5	92.7	22.6	Saturated Steam.
	103.8	26.4	3	55.6	22.7	
	107.4	26.8	2	35.0	24.7	
	106.7	27.9	1	15.5	27.8	
				Brake H. P.	Lb. of steam per Brake H. P. per hour.	
150 H. P. Turbine Motor.	113.8	26.4	7	163.0	17.6	Saturated Steam.
	116.9	25.9	6	138.4	18.2	
	113.8	26.2	5	114.5	17.9	
	114.3	26.5	4	88.3	18.7	
	112.4	27.0	3	64.1	19.0	
	116.2	25.7	2	37.5	22.3	
300 H. P. Turbine Motor.	192.7	27.3	7	303.6	14.1	Steam Super-heated 60 degrees F.
	196.3	27.6	6	255.5	14.7	
	196.3	27.6	5	216.9	14.4	
	196.3	27.6	4	172.6	14.5	
	190.6	27.8	3	121.6	14.9	
	196.3	28.1	2	74.2	17.2	
	413.3	28.5	1	31.5	21.6	

*From paper by Konrad Anderson in *Trans. Inst. Eng. and Shipbuilders in Scotland*, Nov., 1902.

TABLE VIII.

TESTS ON 300 H. P. DE LAVAL TURBINE DYNAMO. (DEAN AND MAIN.)

Number.	Pressures Lb. sq. in. gauge. Above Governor Valve.	Pressures Lb. sq. in. gauge. Below Governor Valve.	Superheat, Degrees F.	Vacuum, Inches.	Brake Horse Power.	Steam Used per Hour. Total Pounds.	Steam Used per Hour. Pounds per B. H. P.
	Tests with Saturated Steam—Average Results.						
1	206.4	196.9	....	26.6	333	5052	15.17
2	207.3	196.5	....	26.8	284.8	4430	15.56
3	207.6	195.8	....	27.85	195.2	3229	16.54
4	201.5	197.9	....	28.1	118.9	1950	16.40
	Tests with Superheated Steam—Average Results.						
5	207.0	198.5	84.0	27.2	352.0	4906	13.94
6	207.4	197.0	64.0	27.4	298.4	4282	14.35
7	200.7	196.6	10.0	27.5	196.5	3033	15.44
8	202.4	197.7	16.0	27.4	196.5	3062	15.62
9	Average of 7 and 8.						15.53

TABLE IX.

SUMMARY OF DEAN AND MAIN TESTS ON 300 H. P. DE LAVAL TURBINE.

Relative Steam Consumption at Different Loads.

Group No.	Loads B. H. P.	Relative Loads.	Steam per Brake Horse Power.	Increase for Diminishing Loads, referred to Maximum Load.
		Saturated Steam.		
1	333	100%	15.17 lb.	
2	285	86%	15.56 "	2.6%
3	195	59%	16.54 "	9.0%
4	119	36%	16.40 "	8.1%
		Superheated Steam		
5	352	100%	13.94 lb.	
6	298	85%	14.35 "	2.9%
9	196	5[illegible]%	15.53 "	11.4%

Saving (at the Turbine) by the Use of Superheated Steam.

	Amount of Superheat.	Load with Superheated Steam.	Load with Saturated Steam.	Steam used per Brake H. P. with Sup. Steam.	Dry Steam used per Brake H. P. with Sat. Steam.	Saving by use of Superheated Steam.
1 and 5	84° F.	352 H. P.	333 H. P.	13.94 lb.	15.17 lb	8.8%
2 and 6	64° F.	298 "	285 "	14.35 "	15.56 "	8.4%

TABLE X.

MISCELLANEOUS TESTS ON PARSONS TURBINES.*

At Stop Valve: Pressure Above Atmosphere, Lbs. per sq. in.	At Stop Valve: Superheat, Degrees F.	Vacuum, Inches.	Speed, Rev. per Minute.	Load in Kw.	Steam Used per Hour: Total Pounds.	Steam Used per Hour: Pounds per Kw.
Test No. 1. 75-Kw. Continuous-Current Turbo for Banbury.						
141.2	84.2	27.1	4,140	75.7	2,006	26.4
141	0	27.0	4,140	75.2	2,201	29.2
142	0	27.1	4,140	56.6	1,777	31.2
Test No. 2. 200-Kw. Continuous-Current Turbo for Shipley						
150	57	27	3,000	204.2	4,538	22.23
151	55	27.9	3,000	101.2	2,696	26.67
156	181	27.3	3,000	202.5	4,130	20.39
151	166	28.0	3,000	100.27	2,446	24.41
Test No. 3. 375-Kw. Turbo-Alternator for Dundee.						
152.9	.....	27.4	3,000	376.9	8,143	21.6
149.4	148.9	27.5	3,000	374.06	7,202	19.25
Test No. 4. 300-Kw. Turbo-Alternator.—Hulton Colliery.						
158.0	0	15.33	3,000	297.4	8,732	29.36
157.0	0	19.33	3,000	305.1	8,369	27.43
152.0	0	22.33	3,000	303.4	7,764	25.59
154.0	0	25.33	3,000	303.15	7,336	24.19
158.0	0	26.58	3,000	303.2	7,020	23.15
Test No. 5. 300-Kw. Turbo-Alternator.—De Beers Explosive Works.						
150.0	53.3	27.88	3,000	312.1	6,260	20.06
153.0	50.0	27.78	3,000	231.8	4,960	21.45
150.5	40.2	27.9	3,000	154.5	3,670	23.75
Test No. 6. 1,500-Kw. Turbo-Alternator.—Newcastle-on-Tyne E.S. Co.						
196	76	27.45	1,200	1,442	25,962	18.0
197	84	27.85	1,200	1,015.5	20,124	19.8
196	76	27.95	1,200	714.0	15,288	21.4
199	77	28.35	1,200	360.5	9,114	25.2
200	68	28.45	1,200	......	2,948	....
After 16 months' use the following figures were obtained:						
203	92	26.11	1,210	1,823	32,431	17.7
207	66	26.46	1,208	1,513	27,582	18.23
Test No. 7. 1,500-Kw. Turbo-Alternator for Sheffield Corporation.						
With vacuum augmentor, and including 450 lbs. steam per hour used by it.						
113.6	108.3	26.69	1,455	1,316.5	24,732	18.76
111.6	156.4	27.12	1,500	1,061.6	19,830	18.66
141	113	27.72	1,500	512.7	11,425	22.3
154	47.5	27.72	1,500	0	3,128	0

*From a paper by Hon. Chas. A. Parsons, G. Gerald Storey, and C. P. Martin, presented before the British Institute of Electrical Engineers, May, 1904. In summarizing the tests of this table, and other tests upon his turbines, Mr. Parsons states: "It will be seen that under conditions of, say, 110 pounds steam pressure and 100 degrees superheat, and a vacuum of 27 inches, the consumptions in round numbers are as follows: A 100 Kw. plant takes about 25 pounds of steam per Kw. hour at full load; a 200 Kw. takes 22 pounds; a 500 Kw. 20 pounds; a 1,000 Kw. 19 pounds; a 1,500 Kw. 18 pounds; and a 3,000 Kw. 16 pounds. These figures are derived from averages of a large number of tests that have been made from time to time. Without superheat the consumptions are about 10 per cent more."

Table X. (Continued).

At Stop Valve.					Steam Used per Hour.	
Pressure Above Atmosphere, lb. per sq. in.	Superheat, Degrees F.	Vacuum, Inches.	Speed, Rev. per Minute.	Load in Kw.	Total Pounds.	Pounds per Kw.
Without vacuum augmentor.						
115.6	143	25.18	1,500	1,039.3	21,261	20.7
137	119	25.97	1,500	534.25	12,820	24.08
150.8	72.4	25.62	1,500	0	2,957.4	0
3,000-Kw. Parsons Turbo-Alternator Supplied to Frankfort by Messrs. Brown, Boveri & Co.						
138.5	235	27	1,350	2,998	44,200	14.74
170.5	187	27.5	1,350	2,518	38,300	15.59
142	120	27.2	1,350	2,600	41,200	15.8
139	114	27.2	1,350	2,600	41,400	15.9
168.5	184	27.9	1,350	1,945	30,800	15.84
146	120	27.6	1,350	2,000	32,600	16.3
157	101	27.4	1,350	1,442	25,400	17.6
142	30	29.3	1,350	0	4,700	excited
142	30	29.3	1,350	0	3,560	non-excited

Table No. XI.

Tests on Parsons Turbines, When Running Non-Condensing.*

At Stop Valve.						Steam Used per Hour.	
Pressure Above Atmosphere, Lb per Sq. In	Superheat, Degrees F.	Back Pressure, Lb. per Sq. In. Gauge	Vacuum, Inches.	Speed, Rev. per Minute.	Load in Kw.	Total Pounds	Pounds per Kw.
Test No. 1. 250-Kw. Continuous-Current.—Messrs. Guinness, Son & Co.							
144	0	0	.....	3,047	251.55	9,540	37.80
142.6	0	6	.....	3,047	255.82	10,584	41.38
138	0	11.1	.....	3,055	253.15	11,194	44.15
143	0	11	....	3,115	125.45	7,475	59.58
Test No. 2. 300-Kw. Turbo-Alternator.—Hulton Colliery.							
161	0	....	0	3,040	296.6	10,180	34.2
158	0	....	26.58	3,000	303.2	7,020	23.15
Test No. 3. 500-Kw. Turbo-Alternator.—Metropolitan E. C. Co.							
142	0	....	0	1,800	506.2	16,903	33.39
146	0	....	22.57	1,800	509.85	13,714	26.89
146	0	....	28.	1,800	500	.....	23.5†

*From a paper by Hon. Charles A. Parsons, G. Gerald Storey and C. P. Martin, presented before the British Institute of Electrical Engineers, May, 1904.

†Result estimated by the author from curve plotted in original paper.

TABLE XII.

TESTS ON RATEAU TURBINE OF 500 H. P. LOCATED AT PENARROYA, SPAIN.*

Data.	¼ Load.	½ Load.	Full Load.	Overload.	Overload at increased speed (2400 rev.)
Electrical H. P. at brushes...	135	259	525	627	641
Admission pressure, absolute, lb. per sq. in..................	46.21	76.6	136	156	156
Exhaust pressure, absolute, lb. per sq. in................	1.24	1.33	1.63	1.82	1.82
Theoretical steam consumption of perfect engine per H. P. hour, lb................	10.93	9.8	8.89	8.73	8.73
Actual steam consumption per electrical H. P. hour at brushes, lb..................	21.3	18	15.8	15.39	14.90
Combined efficiency of the electrical generating set referred to the perfect engine	0.513	0.540	0.560	0.569	0.580

TABLE XIII.

TESTS ON A ZOELLY TURBINE OF 500 BRAKE HORSE POWER.†

Test Number.	In Steam Pipe.		After Passing Governor Valve.		Vacuum, Inches.	Net Power in Kilowatts.	Steam Used per Hour.	
	Pressure lb per sq. in. Absolute.	Superheat Degrees F.	Pressure lb. per sq. in. Absolute.	Superheat Degrees F.			Total Pounds.	Pounds per Kw.
	Tests with Dry, Saturated Steam.							
1	158.47	....	143.3	....	....	363.06	7903	21.75
2	158.47	....	143.4	....	28.5	387.65	8337	21.50
3	154.98	....	128.4	....	28.5	334.51	7825	22.21
4	156.34	....	98.2	....	28.4	240.1	5778	24.08
5	155.77	....	77.4	....	28.3	182.22	4683	25.70
6	156.76	....	48.5	....	28.6	80.18	2650	33.08
7	156.62	....	17.3	39.4	28.6	......	1025	.....
8	158.89	....	10.6	53.0	28.5	......	649	.....
	Tests with Superheated Steam.							
9	181.9	185	138.0	102	28.7	391.66	7454	19.08
10	186.44	153	138.0	107	28.7	389.6	7335	18.82
11	159.89	109	139.0	101	28.7	390.4	7730	19.90

*Quoted by Prof. Rateau in various papers, including one presented before the A. S. M. E. in June, 1904.

†From Data furnished the Author by the Builders.

Table XIV.

Tests on 500 Kw. Curtis Turbine, Cork (Ireland) Electric Tramway and Lighting Co.'s Station.*

Initial Pressure lb. sq. in. Gauge.	Super-heat Degrees F.	Vacuum, Inches.	Average Load in Kw.	Approx. Load.	Steam Used per Hour.	
					Total Pounds.	Pounds per Kw.
155	51	28.8	125.87	¼	3253	25.9
155	50	28.6	250.06	½	5600	22.4
153	70	27.8	393.8	¾	8251	20.9
153	104	26.9	511.7	Full	10503	20.5
151	124	26.2	613.5	1¼	12811	20.9

*Electrical Review (English), Nov. 18, 1904.

Table XV.

Tests on 2000 Kw. Curtis Turbine.*

Load in Kw.	Revolutions per Minute.	Initial Pressure, Gauge.	Vacuum, Referred to 30 in. Merc.	Superheat, Deg. F.	Lb. Steam per Kw. Hour.
1970	918	162	28.15	210	15.12
1040	928	167	28.38	190	15.87
560	930	163.5	28.40	210	17.86
2005	918	169	28.87	125	15.80
1066	928	171	28.48	105	16.33
1970	918	165	28.21	105	16.20
0	932	165.5	28.05	157	Steam per Hour. 1530

*Test by A. R. Dodge, Schenectady, N. Y. Reported by August H. Kruesi, in [illegible] paper before the National Electric Light Association, June, 1905.

TABLE XVI.

TESTS ON 400 KW. (580 H. P.) WESTINGHOUSE-PARSONS TURBINE.*

	Initial Pressure, lb. sq. in. Gauge.	Super-heat, Degrees F.	Vacuum, Inches.	Brake Horse Power.	Steam Used per Hour. Total Pounds.	Steam Used per Hour. Pounds per B. H. P.
Results with Approximately 100° F. Superheat.						
31% Overload	150	104	27.10	758.9	9157	12.07
Full Load (2% Overload)	156	109	27.06	594.6	7384	12.41
77% Load	154	104	27.10	445.8	5728	12.86
41% Load	153	87	27.10	239.9	3503	14.62
Results with Approximately 180° F. Superheat.						
32% Overload	151	182	27.00	762.6	8520	11.17
Full Load (2% Overload)	154	181	27.10	592.3	6779	11.45
Results with Dry, Saturated Steam.						
26% Overload	153	....	26.87	728.4	9928	13.63
Full Load (2% Overload)	154	....	26.84	593.2	8249	13.91
77% Load	156	....	26.80	448.0	6486	14.48
42% Load	156	....	26.90	241.3	3876	16.06
Results with Dry Steam and Poor Vacuum.						
20% Overload	152	....	25.90	694.8	9734	14.01
Full Load (2% Overload)	155	....	25.91	593.1	8514	14.35

With 100° Superheat the speed of rotation was 3,547 R. P. M. at full load; at 31% overload it dropped 2.5% and at 41% load it increased 1.2%.

With Dry Steam the speed was 3,545 R. P. M. at full load; at 26% overload it dropped 1.8% and at 42% load it increased 1.6%. With turbine running light speed increased 8.4%.

*Tests made at the works of the builders by Dean and Main in 1903.

TABLE XVII.

TESTS ON A 1250 KW. TURBINE FOR INTERBOROUGH RAPID TRANSIT CO., NEW YORK.*

	Initial Pressure, lb. sq. in. Gauge.	Superheat, Degrees F.	Vacuum, Inches.	Load Carried by Turbine. In Kilowatts.	In Electrical Horse Power.	In Terms of Full Load.	Steam Used per Hour. Total Pounds.	Lb. per Kw.	Lb. per Electrical H. P.
	Results with Dry, Saturated Steam.								
27" Vacuum	150.3	....	27.08	196.95	264.0	0.157	7155.0	36.33	27.08
	151.4	....	27.11	343.73	459.42	0.276	9738.0	28.4	21.18
	146.8	....	27.1	655.98	879.33	0.525	15074.0	22.98	17.14
	146.5	....	27.11	949.53	1286.5	0.790	20254.0	20.46	15.27
	147.1	....	27.11	1321.46	1771.4	1.060	25712.0	19.47	14.58
	148.0	....	27.05	1489.4	1996.5	1.190	28208.0	18.95	14.13
	144.5	....	27.05	1713.5	2297.1	1.370	33036.0	19.28	14.38
	141.8	....	26.79	1988.9	2666.0	1.590	40547.0	20.39	15.21
28" Vacuum	151.0	....	28.05	334.78	448.74	0.268	9295.0	27.78	20.71
	146.8	....	28.1	972.0	1303.0	0.778	19434.0	19.99	14.91
	146.1	....	28.08	1363.95	1828.3	1.091	25639.0	18.70	14.03
	Results with 75° F. Superheat.								
27" Vacuum	151.8	76.6	27.15	191.0	256.0	0.153	6734.0	35.25	26.3
	151.9	77.2	27.07	333.55	447.1	0.267	9170.0	27.58	20.51
	150.0	77.0	27.15	664.67	891.0	0.531	14181.0	21.33	15.9
	147.7	76.15	27.1	986.23	1322.04	0.790	19108.0	19.39	14.45
	146.3	76.0	27.1	1293.9	1734.9	1.038	23908.0	18.48	13.78
28" Vacuum	151.8	75.5	28.05	198.4	266.1	0.150	6300.0	31.76	23.68
	150.8	77.4	28.05	333.15	446.6	0.266	8430.0	25.30	18.86
	147.6	76.5	28.1	977.64	1310.5	0.780	18180.0	18.59	13.87
	146.0	78.25	28.1	1274.2	1708.0	1.020	22504.0	17.66	13.17

Tests upon a 500 Kw. Curtis Turbine at Newport, R. I.—A Curtis turbine at the Newport, R. I., power house of the Old Colony Street Railway Company, a description of which plant will be given later, was tested by George H. Barrus, consulting engineer, Boston, Mass.† This turbine is one of the earlier two-stage type and had been in continuous service nearly a year when the test was made. The results with dry, saturated steam, 150 pounds initial pressure, and two inches back pressure in the condenser, are tabulated herewith for different loads, together

*Tests made at the Westinghouse Machine Company's Plant, and reported by A. M. Mattice, Chief Engineer

†*The Iron Age*, May 5, 1904.

with some general results with both saturated and superheated steam.

At full load,	19.78	pounds
At three-fourths load,	20.69	"
At half load,	21.38	"
At one-fourth load,	27.85	"
At 50 per cent overload,	20.22	"

With the variable commercial load at the station, which ranges from 333 Kw. to 114 Kw. and averaged 253.2 Kw., the consumption of dry steam was 22.38 pounds per Kw. hour. When the commercial load was augmented by a constant rheostat load, bringing the average up to 421.9 Kw., the steam consumption was 20.7 pounds per Kw. hour.

With superheating of 150.5 degrees at the throttle valve, the steam consumption was 17.79 pounds per Kw. hour at full load, or 10 per cent less than with saturated steam. When superheating was increased to 289.6 degrees, the consumption was 15.1 pounds per Kw. hour, or 19.6 per cent less than with saturated steam. In these tests the load on the auxiliaries averaged about 16.7 Kw., or about 3 per cent of the normal rating of the turbine.

Best Turbine Results.—Table XVIII. summarizes the best results of tests upon turbines quoted in this chapter. With the exception of tests upon two turbines, these results are all in terms of kilowatts or electrical horse-power, and before any comparisons can be made with piston engines, there must be allowances for efficiency, as explained in the first part of the chapter. To facilitate comparisons, certain efficiencies have been assumed, as given in the last column of the table; and by using these efficiencies the equivalent rates of consumption per indicated horse-power per hour were calculated and tabulated in Column 6. The efficiencies were chosen on the following basis:

Combined efficiency of generator and engine in units of from 400 to 500 H. P. (or 300 to 400 Kw.), 88 per cent.

Combined efficiency of units of from 500 to 1,500 Kw., 90 per cent.

Combined efficiency of 2,000 to 3,000 Kw. units, 92 per cent.

Table XVIII.

Examples of Steam Consumption of Turbines. Best Results of Tests Quoted in This Chapter.

Turbine.	Nominal Power.	Steam Used per Hour. Pounds per B. H. P.	Pounds per E. H. P.	Pounds per Kw.	Estimated Equival'nt Consumption per I. H. P.	Per cent efficiency assumed in estimating I. H. P. Results.
1	2	3	4	5	6	7
Results with Saturated Steam.						
De Laval..............	300 H. P.	15.17	.....	.....	13.96	92
Rateau.................	500 H. P.	.....	14.90	.....	13.11	88
Zoelly..................	500 H. P.	.....	16.05	21.50	14.12	88
Curtis (American).......	500 Kw.	.....	14.76	19.78	13.28	90
Westinghouse-Parsons	400 Kw.	13.63	.....	.....	12.68	93
Westinghouse-Parsons	1250 Kw.	.....	14.13	18.95	12.72	90
Results with Superheated Steam (Moderate Superheat, not exceeding 150°).						
De Laval	300 H. P.	13.94	.....	.....	12.82	92
Zoelly..................	500 H. P.	.....	14.05	18.82	12.36	88
Curtis (English)........	500 Kw.	.....	15.29	20.50	13.76	90
Curtis (American)......	500 Kw.	.....	13.28	17.79	11.95	90
Parsons................	300 Kw.	.....	14.96	20.06	13.16	88
Parsons................	1500 Kw.	.....	13.44	18.0	12.10	90
Parsons................	3000 Kw.	.....	11.79	15.8	10 85	92
Westinghouse-Parsons	400 Kw.	12.07	.....	.....	11.23	93
Westinghouse-Parsons	1250 Kw.	.....	13.78	18.48	12.40	90
Results with Highly Superheated Steam (Superheat from 180° to 290°).						
Curtis (American)......	500 Kw.	.....	11.26	15.1	10.14	90
Curtis (American)......	2000 Kw.	.....	11.27	15.12	10.36	92
Parsons................	3000 Kw.	.....	11.00	14.74	10.12	92
Westinghouse-Parsons	400 Kw.	11.17	.....	.....	10.39	93

Engine efficiency alone (without generator) for units of from 400 to 500 H. P (or 300 to 400 Kw.), 93 per cent.

Engine efficiency corresponding to the 300 H. P. De Laval turbine, 92 per cent.

Chart for Estimating Rate of Steam Consumption.—If results on any other efficiency basis are desired, they may be easily calculated, or they may be obtained by the aid of the accompanying chart, Fig. 1. On this chart the figures at the left are pounds of steam per electrical horse-power per hour. The inclined lines are for various efficiencies, and at the top are corresponding values for pounds of steam per indicated or brake horse-power per hour. Obviously, also, if brake horse-power units are assumed to be at

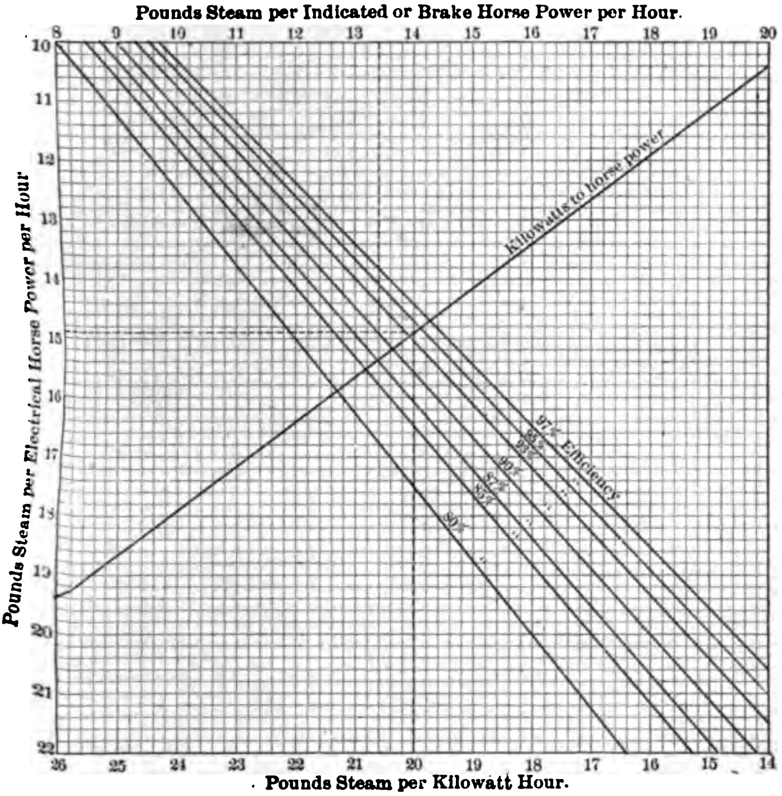


Fig. 1. Chart for Estimating Rate of Steam Consumption.

the left, indicated horse-power units corresponding can be found at the top. The inclined line labeled "kilowatts to horse-power" is for the purpose of converting kilowatt units at the bottom to horse-power units at the left.

Example to Illustrate Use of Chart.—Following the dotted line, we find that 20 pounds of steam per kilowatt hour=14.9 pounds per electrical horse-power per hour. Assuming 90 per cent efficiency, the equivalent steam consumption per indicated horse-power per hour is found by retracing the dotted line from the 14.9 point until it meets the 90 per cent line; then extending upward until it reaches the 13.4 point, which gives the required rate of consumption.

Table XIX.

Examples of Tests upon Reciprocating Engines of Exceptionally High Economy, Showing Best Results Obtained.

Engine.	Horse Power.	Steam Pressure, Gauge.	Vacuum, Inches.	Revolutions per Minute.	Superheat, Degrees F.	Steam per H. P. per Hour, Pounds.	Authority.
Westinghouse Vertical at Brooklyn, N. Y....	5400	185	27.3	76	....	11.93	*Eng. Record*, May [illegible] 1904.
Rockwood-Wheelock at Natick, R. I........	595	159	25.4	76.4	....	13	F. W. Dean, *Trans. A. S. M. E.*, 1895.
McIntosh and Seymour at Webster, Mass.....	1076	123	27.10	99.6	20	12.76	F. W. Dean, *Trans. A. S. M. E.*, 1898.
Rice and Sargent at Brooklyn...............	627	151	28.6	121	....	12.10	D. W. Jacobus, *Trans. A. S. M. E.*, 1903.
Rice and Sargent at Philadelphia..........	420	142	25.8	102	297	9.56	D. W. Jacobus, *Trans. A. S. M. E.*, 1904.
Horizontal, Four-valve	658	150.4	26.4	80	16.4	12.03	Barrus' *Engine Tests*.
Leavitt Pumping Engine at Chestnut Hill, Mass..................	575.7	175.7	27.25	50.6	....	11.2	E. F. Miller in *Technology Quarterly*, Vol. IX.

Best Reciprocating Engine Performance.—In Table XIX. are a few "best results" selected from tests upon several very economical engines. Many other results as good as these could have been tabulated, but the ones given are indicative of what is now attained under the most favorable conditions. It is conservative to say that compound engines may now be built to produce an indicated horse-power on 12.5 pounds of steam per hour, with saturated steam. With a high degree of superheat the long sought 10-pound mark has nominally been passed, but if the results were recalculated in terms of the equivalent rate of consumption of saturated steam, by using the heat unit method, it would be found that they barely reached 10 pounds.

Average Engine Performance.—Since the results in Table XIX are from picked tests, and are exceptional, Table XX. is given which fairly represents what the ordinary high-grade engine will do. This table is made up from the results of tests upon four-valve, compound condensing engines, published in Barrus' *Engine Tests*. They are not selected tests, but are from 23 engines in commercial operation and are average representatives of their class. We find that one of the results falls below 12 pounds, five

Table XX.

Results of Tests on Four-valve Engines—the First Five Engines Operated with Steam Slightly Superheated (12 to 44.5 Degrees); the Others with Saturated Steam

Indicated Horse Power.	Steam per I. H. P. per Hour, Pounds.	Indicated Horse Power.	Steam per I. H. P. per Hour, Pounds.
659	11.89	725	13.27
658	12.08	1714	13.27
670	12.29	1080	13.21
798	13.28	843	13.53
1017	13.26	382	14.05
689	12.69	373	14.18
708	12.45	676	14.6
636	13.28	1540	14.1
[illegible]	13.37	800	15.78
719	13.09	606	16.28
741	13.23	716	19.36
739	13.01		

results fall below 13 pounds, and 16, or over two thirds, fall below 14 pounds, while only three are above 14 pounds. This table confirms what is commonly accepted among engineers; viz., that a rate of consumption of from 13 to 14 pounds is to be expected with engines of this class. A figure of 13.5 pounds may be set as the rate for large engines, operating under good conditions, with saturated, or slightly superheated steam.

Table XXI.

Tests on Compound Engine at Ghent,* Belgium, by Prof. Schröeter, Showing Effect on Economy of Varying Superheat.

Indicated Horse Power.	Superheat, Degrees F.	Steam per I. H. P. per Hour, Pounds	Equivalent Pounds of Saturated Steam.	Heat Units per I. H. P. per Hour.
222	0	12.08	12.08	250
226	43.7	11.58	11.77	244
227	97.7	11.00	11.41	237
223	151.7	10.67	11.33	234
223	221.2	9.81	10.69	221
218	310.9	8.89	10.01	207

Effect of Superheat on Steam Engine Economy.—Since turbine tests are nearly all made with superheated steam, and engine tests almost invariably with saturated steam, it will be useful to

*See paper by Prof. Storm Bull, *Journal of the Western Society of Engineers*, December 1, 1903.

hav[illegible] [illegible]neans for estimating the steam consumption of engi[illegible] [illegible]he supposition that superheated steam is used. Data fo[illegible] [illegible]fforded by tests upon a Belgian engine of 250 horsepower, [illegible]h has established a remarkable record for economy. The tests are summarized in Table XXI., and undoubtedly are reliable, as they were made by Professor Schröeter, one of the most expe[illegible]nced experimenters abroad. The tests start with saturated steam and show the extremely low steam consumption, for an engine of this size, of 12.08 pounds per indicated horsepower hour. The [illegible] [illegible]his one give the consumption for different degrees of [illegible] The results show that for every 100 degrees superh[illegible] [illegible]eam consumption per horsepower per hour was redu[illegible] [illegible] pound, or 8.5 per cent; and that the consumption, ex[illegible] in terms of equivalent consumption of saturated steam, was reduced $\frac{9}{10}$ pound.

Comparing Turbine and En[illegible] Results.—The reader has at his disposal in the last three tables, together with Table XVIII., sufficient information to form an opinion upon the comparative rate of steam consumption of turbines and engines when operating at their most economical loads. Due allowances, however, must be made for sizes of machines, conditions of operation, such as steam pressure, vacuum, superheat, etc. This is very important, as entirely erroneous opinions are often formed where such allowances are not made. Taking 12.5 pounds, previously mentioned, as a conservative figure for the most economical engines using saturated steam; and 13.5 pounds as a safe figure for the *average* high-grade engine, we should then have the estimated rate of consumption for each, with different degrees of superheat, as follows, taking the Belgian figures as a basis:

Most Economical Engine.

With Saturated Steam.	100 Degrees Superheat.	200 Degrees Superheat.	300 Degrees Superheat.
12.5	11.4	10.4	9.3

Average High-Grade Engine.

13.5	12.4	11.2	10.1

Comparing these figures with those of Table XVIII., it seems probable that the reciprocating engine will, under exceptionally good conditions, show a little better economy than the turbine, when running at its most economical load; but that what we have called the "average high-grade engine" appears to about equal the turbine in its rate of steam consumption at most economical loads. In the next chapter the question of variable loads will be considered.

Economy of Small Engines and Turbines.—The author has several times seen it stated by English engineers that in sizes of 500 Kw. and less the engine is more economical than the turbine, but that as sizes increase the economy of the turbine improves more rapidly than that of the engine, and in the larger powers the turbine is equal or superior to the engine.* This view seems to be borne out by the facts if the four-valve compound type of engine be taken for comparison. Engines of this type of from 300 to 500 Kw. capacity are exceptionally economical and are reasonably so in still smaller sizes. The same is true of the Willans engine, used so extensively in England. When we come to the single-valve, high-speed engine, however, which is in such general use in this country, there is no doubt that its rate of steam consumption can easily be improved upon by the turbine. The General Electric Company, the builders of the Curtis turbine, recognize this fact and do not provide for as complete expansion of the steam in turbines of small sizes as in their larger machines, since it is not necessary to do so in order to compete with the high-speed engine.

*In discussing the paper by Parsons, Storey and Martin, before the Institution of Electrical Engineers, May, 1904, E. J. Fox said: "Taking figures of steam economy as given by Mr. Parsons, for different sizes of turbines, there is no difficulty whatsoever in the reciprocating engine giving equally good results up to the 1,500 Kw. size. From 100 up to 1,000 Kw., the results obtainable with reciprocating engines are better. When you come to the 1,500 Kw. size, there is very little difference between the two; and finally with the 3,000 Kw. size, I think there is no doubt a considerable difference in favor of the turbine."

CHAPTER X

STEAM TURBINE PERFORMANCE (Continued).

Characteristics of Turbines Under Variable Loads.

In a steam engine, and in certain turbines, like the Parsons, in which latter there is an auxiliary valve to admit steam to the low-pressure end in case of heavy overloads, the lowest steam con-

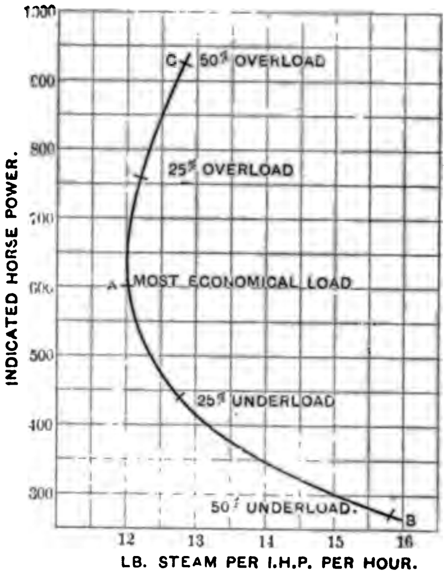


Fig. 1. Corliss Engine Curve.

sumption per horse-power hour occurs at or near the normal load. A decrease in the load below its normal point causes an increase in the rate of consumption; and an increase in the load above the normal point produces a like effect, though to a less degree.

In Fig. 1 is a steam consumption curve for a Corliss engine. The most economical load is at point *A*, situated on the curve at the extreme left, and points *C* and *B*, above and below *A*, respectively, are both to the right of *A*. These two extreme points represent the consumption at 50 per cent overload and underload,

respectively, and it is evident that a greater range of loads is possible than if, for example, the maximum load were at point *A*, where the highest economy occurs, and the consumption at other loads were represented entirely by that part of the curve extending from *A* to *B*.

This latter condition exists in any turbine in which the speed

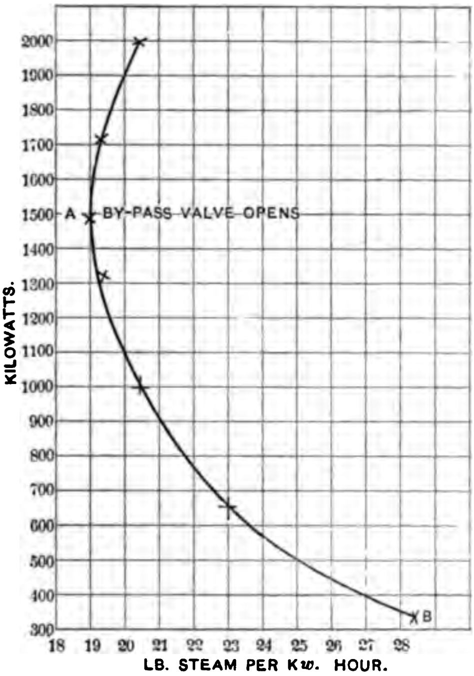


Fig. 2. Parsons Turbine Curve.

is regulated entirely by throttling the steam, as examination of the diagrams shortly to be referred to will make evident.

Curve for the Parsons Turbine.—In Fig. 2 is a diagram showing graphically the rate of steam consumption of the 1,250 Kw. Westinghouse-Parsons turbine, tests on which were summarized in Chapter IX. The curve in this diagram resembles that in Fig. 1 for the Corliss engine. The point where the by-pass opens is clearly defined, and it is the action of this by-pass which gives the resemblance in this curve to the steam-engine curve. If there

were no by-pass the turbine would not be able to carry a load above 1,500 Kw., and we should have a curve extending from *A* to *B*, simply, like the lower part of the engine curve. The governor in this type of turbine is virtually a throttling governor and the rate of steam consumption gradually decreases, as the load increases, until the by-pass opens, when the rate increases, since the steam which enters the low-pressure end through this valve is not used to so good advantage.

Curves for Turbines of the Rateau Type.—In Fig. 3 is a steam rate curve for a Rateau turbine, plotted from Table XII., Chapter IX. This turbine, also, is regulated by a throttling governor and the curve resembles the sections from *A* to *B* of the Corliss engine, and the Parsons curves, Figs. 1 and 2. The most

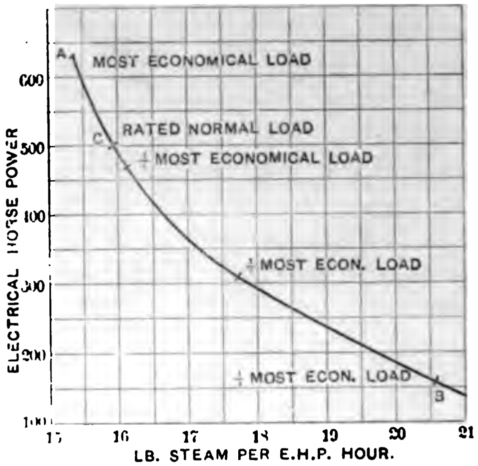


Fig. 3. Rateau Turbine Curve.

economical, and also the maximum, load carried by the Rateau turbine is at point *A*, at the upper end of the curve, Fig. 3. The normal load is at point *C*, and the fractional loads are at the successive points indicated. Turbines of this type cannot be strictly said to have overload capacity, and their normal rated load must be fixed at some point below the most economical load, in order to give the machine the equivalent of overload capacity. Professor Rateau, however, has proposed the use of a by-pass valve,

in which case the performance in respect to variable loads would be substantially the same as in the Parsons type fitted with this device.

In Fig. 4 are three curves plotted from tests in Chapter IX. on the 400 Kw. Westinghouse-Parsons turbine, given in Table XVI., the Rateau turbine, Table XII., and the Zoelly turbine, Table XIII. The curves are placed in their correct relative positions so that comparisons can be made. It should be noted that

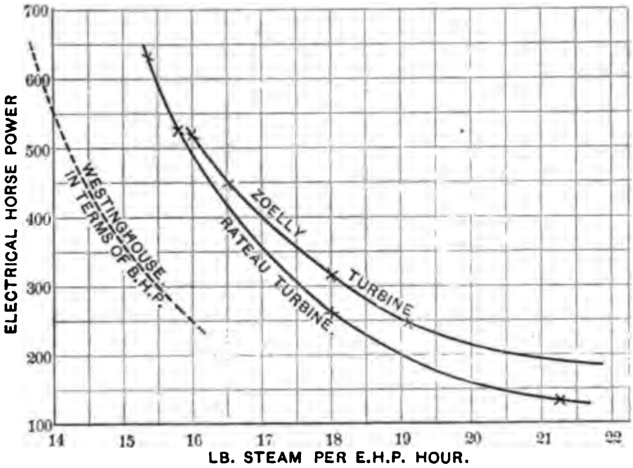


Fig. 4.

the Rateau and Zoelly curves are based on the electric horsepower and are modified somewhat by the efficiency of the generator, while the Westinghouse curve is based on brake horsepower. The similarity of the curves, however, will be apparent.

Curve for the Curtis Turbine.—This turbine operates under boiler-pressure steam at all loads, the governor changing the nozzle area instead of reducing the pressure in the steam chest, as in the other types mentioned. In Fig. 5 is a curve plotted from the Curtis turbine tests of Table XIV. The least steam consumption occurs at the normal load of 500 Kw., and above normal load the curve bears to the right to point *a*, just as in the steam-engine curve, Fig. 1; though theoretically it should continue up to point *b*. From ¾ load to 1¼ load the change in steam consumption is

but from ¾ load to ¼ load it is rapid. The increased ons at light loads in this turbine is due to internal losses, such tion, diffusion and eddying of the jets, radiation, etc., which nearly constant and therefore absorb a larger per- cent the power at light loads than at heavy loads. Com- pari curve with that of the Rateau turbine, we find the lat- ter rese s that part of the Curtis curve lying between a point at abo load and ¼ load. The upper part of the Curtis curve, where re is considerable variation in power with only slight variation in rate tion, is absent in the Rateau

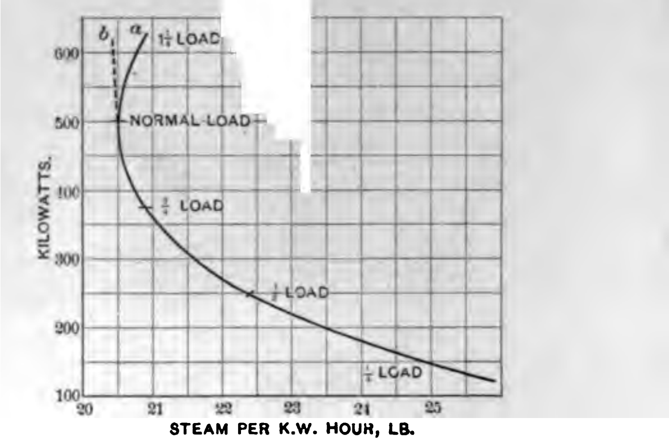


Fig. 5. Curtis Turbine Curve.

curve. This is explained by the fact that in the Rateau, or any turbine which governs by throttling only, there are not only the constant internal losses just mentioned in connection with the Curtis turbine, but there are also losses due to the throttling of the steam, and the two together cause the steam rate to increase more rapidly with a drop in load than in the Curtis turbine, where the steam is not throttled.

Curve for the De Laval Turbine.—In the De Laval turbine conditions are very similar to those found in the Curtis turbine. While regulation for small changes in load is effected automatically by throttling, for wide variations in load the several steam nozzles are opened or closed by hand, as required. Two

curves are plotted in Fig. 6, one for the last series of tests in Table VII. and one for the first series in Table VIII. in Chapter IX. The De Laval curves have the characteristics of the Curtis curve, but in these cases, at least, the De Laval turbine will evidently run at proportionately lighter loads without marked increase in steam consumption.

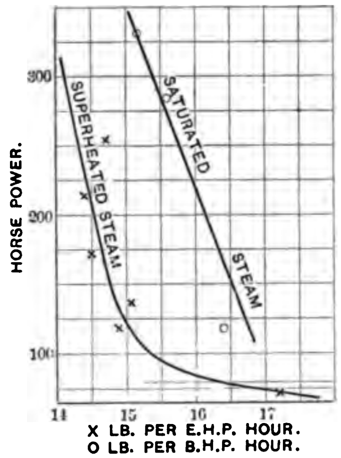


Fig. 6. De Laval Turbine Curves.

Results of Turbine and Engine Tests Under Variable Loads.

Data for Performance under Variable Loads.—Having studied the characteristics of different types of turbines under varying loads, it is now in order to institute comparisons between reciprocating engines and turbines under these conditions. For this purpose figures will be given from the results of a number of tests upon engines of different types. The tests selected for the purpose are those enumerated below, numbered from 1 to 6 for convenience in reference.

1. A slow-running, simple Corliss engine of 150 horse-power at Creusot. Steam pressure, 60 pounds; vacuum, 27 inches; revolutions, 60. Tests reported in Peabody's Thermodynamics.

2. A simple, non-condensing Corliss engine with two 16-inch cylinders, 42 inches stroke. Pressure, 100 pounds; revolutions, 86. Quoted in *Engine Tests* by Geo. H. Barrus.

3. A tandem-compound, condensing Buckeye engine, 160 horse-power.

Cylinders, 11 and 19 by 24; revolutions, 160. Tests given by A. K. Mansfield, Proc. A. S. M. E., 1897.

4. A McEwen, tandem-compound, non-condensing, high-speed engine. Cylinders, 9 and 16 by 14; pressure, 112 pounds; revolutions, 265. Tested by Prof. R. C. Carpenter, Proc. A. S. M. E., 1893.

5. A Fleming, four-valve, tandem-compound, condensing engine, of 500 horse-power. Cylinder ratio, 1 to 7.33; pressure, 150 pounds. Reported by B. T. Allen, Proc. A. S. M. E., 1904. These tests have been criticised because the steam pressure was allowed to drop in the boiler at light loads. It is probable, however, that this variation did not greatly affect the results, since the high-pressure admission valves throttled the steam at light loads, so that full boiler pressure could not have been realized in the cylinder in any case.

6. A Rice and Sargent, compound engine (Corliss type) with cylinders 20 and 40 by 42 inches. Steam pressure, 150 pounds; vacuum, 28 inches; revolutions, 120. Tested by Prof. D. S. Jacobus for the builders.

In obtaining figures from the tests upon the above engines, and also from the turbine tests in this chapter, the method followed has been to plot curves for the rate of steam consumption under varying loads, and from those curves to take the figures at such points as were desired. For example, the results of an engine test might not give the steam consumption at exactly half load, but by plotting the curve for such results as were given, the half-load consumption could be approximately determined. Figures will be given without showing the curves, except where they have previously been printed.*

*The figures given below are the tabulated results from which the curves for the engine tests were plotted. Of each group of figures, the first column contains the indicated horse-powers and the second column the corresponding steam consumptions in pounds per horse-power hour.

Engine No. 1.		*Engine No 2.*		*Engine No. 3.*	
194	18.6	342.4	25.91	200.5	17.55
175	17.7	287.1	25.39	172.	16.3
150	17.3	222.2	25.83	153.7	17.1
117	17.6	146.2	31.43	140.9	17.65
91.7	18.5	100.4	38.38	135.9	17.76
		37.	73.63	103.	18.2
				89.5	18.25
				55.1	24.1

Engine No. 4.		*Engine No. 5.*		*Engine No. 6.*	
126.	21.2	553.49	12.73	1,004.3	12.75
118.4	20.3	501.55	12.66	853.3	12.33
100.6	21.7	348.28	12.33	819.6	12.55
95.1	19.1	321.54	13.59	627.4	12.10
80.5	18.9	87.07	14.42	491.4	13.92
61.	19.69			339.7	14.58
44.	20.7				
27.2	23.11				

The turbine tests selected for comparison are the following, taken from data in Chapter IX.:—

No. 1. 300 H. P. De Laval. Fourth test in Table VII., with superheated steam. Results in terms of electrical horse-power.

No. 2. 300 H. P. De Laval. First test in Table VIII., with saturated steam. Results in terms of brake horse-power.

No. 3. 500 H. P. Rateau. Table XII. Results in electrical horse-power units.

No. 4. 500 H. P. Zoelly. Table XIII. The results in the table were converted into terms of electrical horse-power before plotting curve.

No. 5. 500 Kw. Curtis. Table XIV. Results in pounds per kilowatt hour.

No. 6. 500 Kw. Curtis turbine at Newport, R. I. The figures for this machine were taken from a curve plotted from the results of tests by Geo. H. Barrus and given in a paper by W. L. R. Emmet before the Engineers' Club of Philadelphia, in March, 1904.

No. 7. 1,250 Kw. Westinghouse-Parsons, with by-pass valve. Table XVII. Results in terms of kilowatts.

No. 8. 400 Kw. Westinghouse-Parsons. First and third tests, Table XVI., one with superheated steam and one with saturated steam. Results in terms of brake horse-power. This turbine had no by-pass.

Comparison of Tests under Variable Loads.—One way of comparing tests under variable loads is to set a percentage limit for the rate of steam consumption and then determine how great a variation in load the engine or turbine will permit without exceeding this limit.

The author has assumed a limit of 10 per cent increase in the rate of steam consumption above the most economical rate, and then determined the approximate variation in power for each turbine or engine, corresponding to the 10 per cent variation in the steam rate.

The variation in power was found in per cent of the maximum power developed by each turbine or engine, and is as follows:

Turbine	No. 1.	Variation	in	power,	55%	of	maximum	power.
"	No. 2.	"	"	"	60%	"	"	"
"	No. 3.	"	"	"	35%	"	"	"
"	No. 4.	"	"	"	35%	"	"	"
"	No. 5.	"	"	"	60%	"	"	"
"	No. 6.	"	"	"	60%	"	"	"
"	No. 7.	"	"	"	55%	"	"	"
"	No. 8.	"	"	"	50%	"	"	"

Chapter IX.* Fortunately the tests upon this engine included a record of the electrical output, from which the rate of consumption per electrical horse-power hour could be calculated, enabling direct comparisons to be made with turbines without having to allow for the efficiency of the apparatus.

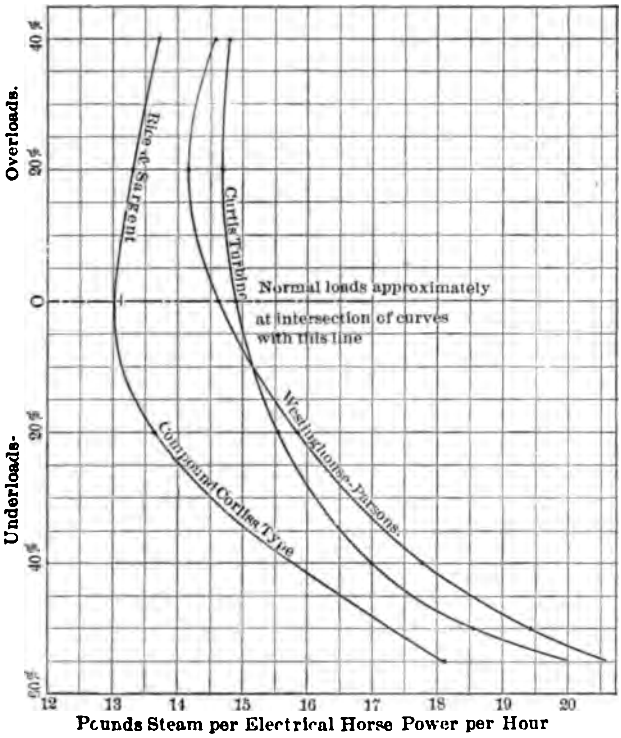


Fig. 7.

*The curve for the Curtis turbine was plotted from a curve showing the results of Barrus tests, given by W. L. R. Emmet of the General Electric Company, in a per presented before the Engineers' Club of Philadelphia, March, 1904. The curve r the Rice and Sargent engine was plotted from a report of Professor Jacobus' tests on this engine, issued by the builders in pamphlet form. The figures are as follows:

Electrical Horse Power.	*Total Weight Steam per Hour.*	*Pounds Steam per E. H. P. per Hour.*
945.6	12,803	13.54
796.4	10,520	13.21
761.1	10,283	13.51
564.4	7,591	13.45
440.2	6,840	15.54
282.9	4,953	17.51

Deductions from Curves of Fig. 7.—The Curtis turbine produces the flattest curve, while there is not much to choose between the Rice and Sargent and the Westinghouse curves, so far as this feature is concerned. In point of economy, the engine is easily in the lead. This engine, however, has proven itself to be exceptionally economical, while the 500 Kw. Curtis turbine as then constructed had only two stages and would not run with as low steam consumption as the larger, three- or four-stage machines. It would lead to erroneous conclusions to accept this diagram too literally as an example of what may be expected from engines and turbines in general, when operating with variable loads. The diagram is a significant one, however, and inevitably leads to the conclusion that a compound Corliss engine is able to hold its own, in comparison with turbines, provided the variation in load is not over 40 or 50 per cent above or below normal load.

Engine Performance from an Operative Standpoint.—In connection with the above comparisons of turbines and engines under variable loads, the author wishes to point out two important facts: First, that the water rate load curve of an engine does not represent its true performance where the load is rapidly fluctuating. The rapidly changing cylinder conditions that exist when the load is fluctuating lead to increased condensation and consequent waste of steam that does not occur when an engine is under test at certain fixed loads.

Second, that in the tests quoted the overloads were not more than 50 per cent above normal load and in most cases were less than this. In one way this adds to the interest of the comparisons because it shows what turbines will do within ranges of load under which reciprocating engines generally operate; but in another way it is not fair to the turbine, because the practice now is to design turbines to carry much heavier overloads than can the steam engine. It is not usual for engines to run with loads greater than 50 per cent in excess of their normal load, it being advisable to put in a larger engine when this point is reached. Corliss engines fitted with two eccentrics *can* carry overloads as great as 100 per cent, provided the vacuum can be maintained, but not with good economy. The tendency always is to install a steam engine large enough to safely handle the overloads, and then let it

operate at an average load considerably below the normal load for the balance of the time. This is illustrated in a striking manner by tests made by students of Cornell University upon 35 street railroad power plants during a period of 12 years. These results have been gathered by Prof. R. C. Carpenter and grouped according to the type of engine.* There were eight tests of compound condensing engines of the Corliss and similar types, and the main results of these are given in Table I. They show the same characteristics as the others of the 35 tests, not quoted here.

TABLE I.

SUMMARY OF TESTS ON COMPOUND CONDENSING ENGINES OF THE CORLISS AND SIMILAR TYPES, IN STREET RAILWAY PLANTS.

H. P. of Engine.	Steam per 1 H. P. per Hour.	Mean Observed H. P.	Per cent of Mean H. P. to Capacity.
825	22.7	482	58.2
1,000	21.9	277	27.7
1,000	20.0	314	31.4
350	16.64	182	52.2
500	16.90	290	58.
2,000	14.50	814	40.7
200	17.30	145	72.
1,600	20.50	...	....
Average	18.80		48.6

Taking the average results, the average load on these engines was less than half their normal load, and they were running day in and day out on a steam consumption of 18.80 pounds per horsepower hour, instead of the 13 or 14 pounds that similar engines are capable of at normal load, if in good condition.

How the Turbine Improves upon Engine Performance.—What is the answer of the turbine to this condition of affairs? The turbine, first of all, is not subjected to serious losses from internal condensation and under a rapidly fluctuating load should show nearly the same economy as indicated by the water rate load curve. This is of great importance in street railway work.

Second, the results that can be secured with a turbine under heavy overloads are indicated by a test upon a Westinghouse-

*Sibley Journal of Engineering, December, 1904.

Parsons 400 Kw. turbine, by F. P. Sheldon & Co., mechanical engineers, Providence, R. I.

This turbine showed the usual results under loads varying from ½ to 1½ the normal rating; but in addition—and this is the important point—demonstrated its ability to carry an overload of 100 per cent with an increase in the rate of steam consumption of less than 10 per cent.

TABLE II.

PERFORMANCE OF WESTINGHOUSE-PARSONS TURBINE UNDER VARIABLE LOADS, INCLUDING HEAVY OVERLOADS.

Steam Pressure 150 Lb. Absolute; Vacuum 28 Inches.

Load on Turbine.	Steam Consumption—Pounds per B. H. P. per Hour.	
	Saturated Steam.	Steam Superheated 100 Degrees.
½ Load	15.86	14.34
¾ Load	15.06	13.45
Full Load	13.89	12.48
1¼ Load	13.83	12.41
1½ Load	.....	12.79
100 % Overload	15.12	13.55

The meaning of this is that a turbine can be installed with reference to the *average* load that it has to carry, instead of with reference to the *maximum* load, as in the case of the steam engine, and then trust to the by-pass to take care of the overloads. The turbine will then be operating at or near its point of best economy most of the time, and being of smaller power than a steam engine for the same work, will not, at light loads, drop to so small a percentage of the normal load as will the steam engine. The recent practice of encasing the turbine generator and cooling it with forced air circulation makes the generator amply able to handle overloads and the generous condensing systems used with turbines should preclude a serious drop in vacuum under excessive overloads.

Taking into account the operative conditions of engines and turbines as they exist, the author is inclined to the opinion that the turbine will surpass the engine under fluctuating loads. This conclusion applies directly to the Parsons type, fitted with a by-pass valve. Enough data have not been published in respect to

the Curtis and other types to show how the matter stands with them, although the guarantees made by the builders of the Curtis turbine are all that could be desired.

The Effect of Vacuum Upon Economy.

In the operation of a steam engine, it is customary to run with a vacuum of about 26 inches in the condenser. While a higher vacuum is slightly advantageous from the fact that it reduces the back pressure upon the piston, the reciprocating engine is not capable of taking full advantage of the increased vacuum, as explained in the chapter upon condensers. In a turbine, however, steam can easily be expanded to the volume corresponding to the lowest pressure attainable in a condenser, and the effort is made to run with a vacuum of 28 or 29 inches.

We are now concerned with the gain in economy resulting from this higher vacuum, or what is more to the point, with the *loss in economy* due to a reduction in the vacuum below the usual figure of 28 inches at which turbines are designed to run. While nearly all turbine tests are made with a vacuum of 28 or 29 inches, and

TABLE III.

ECONOMY OF 750 KW. WESTINGHOUSE-PARSONS TURBINE WITH 26-INCH AND 28-INCH VACUUM.

Approximate Load in E. H. P.	Pounds Steam per E. H. P. per Hour.		Increase in Steam Consumption Due to Low Vacuum.
	With 28-Inch Vacuum.	With 26-Inch Vacuum.	
Tests with Saturated Steam.			
1,500	13.90	14.73	5.3 per cent
1,100	13.76	15.06	9.5 per cent
775	14.65	16.62	13.4 per cent
500	15.95	18.35	15.0 per cent
Tests with Steam Superheated 150 Degrees.			
1,500	11.50	12.98	12.9 per cent
1,275	11.42	13.05	14.3 per cent
1,025	11.79	13.18	11.8 per cent
525	13.85	16.04	15.8 per cent
Average Results with Saturated Steam...	14.59	16.19	10.9 per cent
Average Results with Superheated Steam	13.81	12.14	13.7 per cent

figures are usually quoted on this basis, it is not always, and we doubt if it is usually, possible to maintain so good a vacuum in commercial operation. Accordingly, what we want to know is, how much the steam consumption of a turbine will be increased by this drop in vacuum.

Turbine Tests with Different Vacuums.—The most complete tests of turbines under different vacuums have been made at the shops of the Westinghouse Machine Company, and are reported in their turbine catalogue.* In Table III. is a summary of tests upon a 750 Kw. turbine compiled from this source, and in Table IV. is a summary of tests upon a 1,250 Kw. unit.

Table III. gives results under different loads for vacuums of 28 and 26 inches, respectively, and with both saturated and superheated steam. It is evident that the increase in steam consumption is more marked at light than at heavy loads, and the average increase for the two inches difference of vacuum is 10.9 per cent with saturated steam and 13.7 per cent with superheated steam.

TABLE IV.

ECONOMY OF 1,250 KW. WESTINGHOUSE–PARSONS TURBINE WITH DIFFERENT VACUUMS. TESTS WITH SATURATED STEAM.

Approximate Load in E. H. P.	Pounds Steam per E. H. P. per Hour. With 28-Inch Vacuum.	With 27-Inch Vacuum.	With 26-Inch Vacuum.	With 25-Inch Vacuum.
2,000	14.73	15.22	15.72	16.32
1,200	15.75	17.06	18.29	18.82
400	21.8	24.35	26.62	28.1
Average Results...	17.43	18.88	20.21	21.08
Differences.........	.....	1.45	1.33	0.87
Per cent Increase for Each inch Drop in Vacuum	.....	8.3	7.1	4.3

Increase in Steam Consumption on account of Drop in Vacuum from 28 to 26 inches:

Load, 2,000 E. H. P., Increase = 6.7 per cent.
Load, 1,200 E. H. P., Increase = 16.1 per cent.
Load, 400 E. H. P., Increase = 22.1 per cent.
Average Increase = 15.9 per cent.

*In a turbine, the benefit derived from a good vacuum is much more than in a reciprocating engine, every one inch of vacuum between 23 inches and 28 inches affecting the consumption on an average about 3 per cent in a 100 Kw., 4 per cent in a 500 Kw., and 5 per cent in a 1,500 Kw. turbine, the effect being more at high vacuum and less at low.—*Hon. Chas. A. Parsons* in a paper before the Institution of Electrical Engineers, May, 1904.

Table IV. gives results under different loads for vacuums varying by inches from 28 to 25 inches. The average increase in steam consumption when the vacuum drops from 28 to 27 inches is 8.3 per cent; from 27 to 26 inches, 7.1 per cent; from 26 to 25 inches, 4.3 per cent, which shows that the drop from 28 to 27 inches affects the results nearly twice as much as the drop from 26 to 25 inches. The average increase, as summarized at the bottom of the table, for a drop in vacuum from 28 to 26 inches, is 15.9 per cent. We see from this, and the previous table that a loss of two inches of vacuum, below 28 inches, causes in these cases an increase in steam consumption of say from 10 to 15 per cent. A turbine, therefore, running on 20 pounds of steam per Kw. hour, with 28 inches vacuum, might be expected to use from 22 to 23 pounds if the vacuum was reduced to 26 inches—quite a likely condition.

The diagram, Fig. 8, was plotted from the tests upon the 1,250

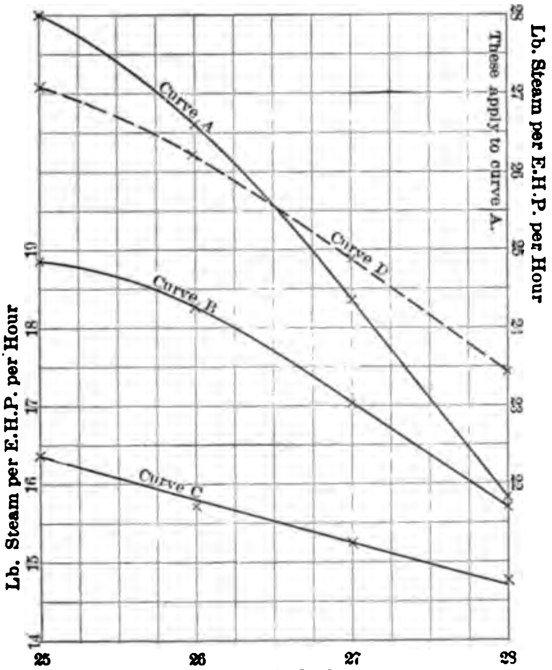


Fig. 8.

H. P. turbine and shows the steam consumption at different vacuums. Curve *A* is for the load of 400 H. P.; curve *B* for 1,200 H. P.; curve *C* for 2,000 H. P.; and the dotted curve *D* for the average of the three loads. This chart shows graphically the two facts already pointed out, that a change in vacuum has a greater effect at light than at heavy loads, and that the effect is more marked at high than at low vacuums.

The Effect of Superheating.

In the tests upon the De Laval and Parsons turbines in the last chapter are data upon the gain (at the turbine) through the use of superheated steam. While such results have little or no significance, unless computed on the heat-unit basis, it is believed to be, and probably is, the case, that superheated steam is desirable for turbine use. In the summary of tests upon a 300-horse-power De Laval turbine in Table IX., the gain from superheating is about 1 per cent for every eight to 10 degrees superheat. In their report upon the tests of a 400 Kw. Westinghouse-Parsons turbine, already referred to, Dean and Main estimate the gain from superheating to be one per cent for each 10 degrees, up to 180 degrees superheat, which was the limit of their tests. This figure is corroborated by Parsons, who, in the paper before the British Institute of Electrical Engineers, from which Tables X. and XI. are quoted, says, "Every 10 degrees superheat up to about 150 degrees F., affects the consumption about 1 per cent."

Reason for Gain from Superheated Steam.—The gain from superheated steam in the turbine comes from reduced friction between the rapidly flowing steam and the passages of the turbine, and between the steam and the surfaces of the rotating wheels. In the steam engine the gain from superheated steam is due to the reduction of cylinder condensation, resulting in less loss from re-evaporation of this moisture at the lower pressures during the latter part of the stroke, and during exhaust. The initial condensation of saturated steam entering an engine cylinder is often as much as 40 or 50 per cent, and this is partly or entirely prevented when the steam is superheated, depending upon the degree of superheat. In the turbine the effect of superheated steam is also to reduce condensation, which always occurs when saturated

steam expands adiabatically, as it is assumed to expand when it flows through a turbine. But the walls of the turbine remain at so uniform a temperature that there is no opportunity for loss through reëvaporation of this condensed steam, and the gain realized must therefore result from the reduced friction, as stated above.

It is fallacious to suppose there is any considerable gain because superheated steam has a higher temperature than saturated steam. The theoretical or thermodynamic gain from superheating is almost nothing. It is true that the heat represented by the superheat is received at a higher mean temperature than the other portions of the heat in the steam, and there is a theoretical advantage from this fact. The amount of heat received at this high temperature, however, is so small, compared with the amount received before superheating begins, as to be almost negligible. One pound of saturated steam at 100 pounds gauge pressure contains nearly 900 latent heat units; and if this steam were then superheated 100 degrees, it would acquire only about six heat units in addition, in spite of its high temperature.

Practical Considerations.—In the account published in *The Iron Age*, May, 1904, of Barrus' tests upon one of the Curtis turbines at Newport, R. I., the reported results with superheated steam were not entirely favorable. This station was equipped with a separately fired Schmidt superheater and unless the superheater was worked to nearly its full capacity there was a loss from its use. This is not to condemn a moderate degree of superheating, however, such as can readily be realized by the use of superheating tubes in the same setting with a steam boiler, and such, moreover, as can be maintained from day to day in actual operation. The author would recommend a turbine plant to be so equipped with capacity for superheating 50 degrees. This ensures perfectly dry steam at the turbine, which is the main point to be attained, without trouble from high temperatures.

Economy with Change in Speed.

The turbine is a one-speed machine. Its speed cannot b changed from that at which it is designed to run without serio loss in efficiency. The speed of the rotating buckets must bear

definite relation to the velocity of the steam, as determined by the proportions of the turbine, or otherwise there will be disastrous impact of the steam against the vanes. In the Curtis turbine at the Newport station, tests were run to determine the variation in steam consumption with a variation in the speed. At 1,900 revolutions per minute, the water rate per kilowatt hour was about 19.75 pounds; at 1,600 revolutions, about 20.75 pounds; at 1,300 revolutions, about 22.7; and at 1,000 revolutions, about 27 pounds. This shows that at first the decrease in speed produced only a small effect on the economy; but later, as the slower speeds are reached, the steam consumption increases very rapidly.

In a paper upon steam turbines* by Ernest N. Jansen, tests are quoted upon several turbines running with varying speeds. With a 400 Kw. Westinghouse-Parsons turbine at 1,800 revolutions per minute, or half of the designed speed, there was an increase of 25 per cent in the steam consumption.

*Published in the *Journal of the Society of Naval Architects*, 1904.

CHAPTER XI

EXPERIMENTS ON THE FLOW OF STEAM.

Napier's Rules for the Flow of Steam.—R. D. Napier was one of the first experimenters to secure results of value on the flow of steam. He published data in 1866 to show that when steam flows through a cylindrical nozzle having a rounded inlet the weight discharged in a given time depends only on the initial absolute pressure, so long as the absolute pressure against which the nozzle discharges does not exceed $6/10$ of that pressure. Thus, if steam flows from a pressure of 100 pounds absolute to a pressure of 60 pounds absolute, the weight discharged in a given time will be practically the same as though the nozzle were discharging at some lower pressure, as into the atmosphere, or into a partial vacuum.

If, however, the final pressure is more than 60 pounds, the weight discharged will be less than before and will become very much less as the difference of pressures decreases.

In the London *Engineer* for November 26 and December 3, 1869, Professor Rankine reviews Napier's work and presents one of the best theoretical discussions of the flow of steam that has been published. He concludes that the formulas given below, commonly known as "Napier's Rules," will give a rough approximation of the weight of steam discharged through a conoidal converging nozzle. The rules also apply in the case of short, cylindrical tubes with rounded inlets.

Let W=flow in pounds per second.

p_1=higher pressure and p_2=lower pressure, both in pounds per square inch absolute, and

a=area of orifice in square inches.

Case I.—Lower absolute pressure equal to or less than $6/10$ of higher absolute pressure:

$$W=\frac{p_1 a}{70} \tag{1}$$

Case II.—Lower absolute pressure more than $\%_{10}$ of higher absolute pressure

$$W = [illegible] \times \sqrt{(p_1-p_2) \div [illegible] p_2} \qquad (2)$$

[illegible]

Case I.

$$W = [illegible] = [illegible] \text{ pounds per second.}$$

Case II. Let [illegible] Then

$$W = [illegible] \times [illegible] \times 80$$

$= [illegible]$

$= [illegible]$ pounds per second.

[illegible] rules give better results in cases where applicable than the more complicated rules based on the laws of thermodynamics.

[illegible].—The next important [illegible] are described in a "Report on Safety Valves," [illegible] in the "Transactions of Engineers and Ship-[illegible] Vol. XVIII. Also contained in London [illegible] December 4 and 11, 1874. Table I. is made up [illegible] sets of tests, in the first of which the higher [illegible] constant and the lower pressure varied; while in the [illegible] pressure was constant and the higher pressure [illegible]

These tests are quoted here because the results illustrate Napier's rules for the flow of steam. In making the tests the weight of flow was measured and the velocity of flow was calculated by theoretical formulas. It will be noted from column 4, of the first group, that the weight of steam discharged increases until the lower pressure drops to [illegible] per cent* of the higher pressure, after which [illegible] constant. This is as it should be from Napier's [illegible] showed that the flow is proportional to the [illegible] the lower pressure is not over $^{6}\!/_{10}$ of the [illegible]

* [illegible] quoted in the table was first used by Weisbach, and is [illegible] as the limiting point at which Napier's rules hold. Both [illegible], however, and vary considerably with the conditions, as [illegible] to this in the first chapter, in connection with the dis-[illegible]

TABLE I.

BROWNLEE'S EXPERIMENTS ON THE FLOW THROUGH CYLINDRICAL ORIFICES.

	Absolute pressure in boiler, lb. sq. in.	Pressure against which nozzle discharged, lb. sq in.	Calculated velocity at throat of nozzle in ft. per sec.	Flow of steam in lbs. per minute per sq. in. of orifice.
	1	2	3	4
First Group. Constant initial pressure. Variable final pressure.	75	74	230	16.68
	75	72	401	28.35
	75	70	521	35.93
	75	65	749	48.38
	75	60	933	56.12
	75	50	1252	64.0
	75	45	1401	65.24
	75	43.46 { 58 per ct. }	1446.5	65.3
	75	15	1446.5	65.3
	75	0	1446.5	65.3
Second Group. Variable initial pressure. Constant final pressure.	165	14.7	1481	140.46
	135	14.7	1472	115.61
	115	14.7	1466	98.76
	90	14.7	1454	77.94
	70	14.7	1441	61.07
	50	14.7	1429	41.06
	40	14.7	1419	35.18
	30	14.7	1408	26.84
	25.37	14.7 { 58 per ct. }	1401	22.81

In the second group the higher pressure drops instead of being constant as before, and as the lower pressure is in each case less than $\frac{6}{10}$ of the higher pressure, the flow should decrease at the same rate that the higher pressure drops, which is actually the case.

Another lesson to be learned from Brownlee's table is that the velocity of flow is practically constant when the lower pressure is $\frac{6}{10}$ or less of the upper pressure. In column 3, the velocity is calculated to be from about 1,400 to 1,480 feet per second when the lower pressure does not exceed $\frac{6}{10}$ of the higher pressure. A value of 1,450 feet is a fair average, and sometimes it is said in round numbers that steam will discharge from either a converging or straight nozzle at the rate of 1,450 feet per second when the lower pressure is $\frac{6}{10}$ or less of the upper.

Flow Through Cylindrical Nozzles.—Experiments were made by L. H. Kunhardt, class of 1889, Massachusetts Institute of Technology, under the direction of Prof. C. H. Peabody upon the flow

of steam through tubes or mouthpieces ¼ inch in diameter. There were three mouthpieces tested having inlets rounded with a radius of 1 inch and straight sections of ¼, ½ and 1½ inches, respectively. The experiments were conducted to find the weight of steam discharged for different differences of pressure. The flow was then calculated by a theoretical formula based on the principles of thermodynamics, and the results compared, which gave the probable coefficient of flow for the three mouthpieces. The flow was also calculated by Napier's formula.

When under test the mouthpieces were screwed into a brass partition between two cast-iron reservoirs, in the first one of which the steam was maintained at a constant initial pressure, and in the second the pressure was varied. The pressure in the tube was found by drilling into the tube at the middle of the straight section of each mouthpiece.

TABLE II.

[illegible] SHORT TUBES WITH ROUNDED INLETS.

[illegible]	[illegible]	[illegible]	[illegible]	Ratio of pressures. Absolute		Flow in pounds per hour.			
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	By experiment.	Calculated by theoretical equation.	Calculated by Napier's empirical equation.	Coefficient of flow: item 7 ÷ item 8.
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

It is to be observed that the actual flow is larger than that calculated by the theoretical equation. This equation is based on the assumption that no heat is lost to or given up by the tube. But it is evident that some heat must have been conducted through the walls of the nozzle from the hot steam in the upper chamber to the steam passing through the tube, and this may explain why the coefficient of flow is greater than unity in each case. The results indicate that this transmission of the heat was more than enough to overcome any loss from friction in the tube.

For the longest tube Napier's rule gives results greater than the actual weight of steam passing through the nozzle, and for the short tubes the results calculated by Napier's rule fall below the actual. This shows the greater effect of friction in the longest tube. The largest discrepancy between the actual results and those given by Napier's formula is about 3 per cent.

The pressure in the tube ranges from .58 to .64 of the upper absolute pressure and is more for the long tube than for the shorter ones, and is also slightly more for the high pressure tests with the short nozzle than for the low pressure tests. The question of the pressure of steam at or near the throat of the nozzle has an important bearing on the design of a nozzle. These tests and others yet to be quoted show that the throat pressure does not vary widely, and that it lies between .5 and .7 of the upper absolute pressure, or at a mean of about .6. Theoretically it should be .58 of the upper pressure.

Experiments on the Discharge of Steam Through Orifices, by Strickland L. Kneass.*

These experiments were completed in 1890 at the works of William Sellers & Co., Inc., Philadelphia, in connection with their steam injector work. Their object was to determine the behavior of steam within a discharging nozzle, and the extent to which the terminal velocity is affected by changes in the proportion of the tube. The nozzles tested were 8mm. (.31496 inch) internal diameter at the throat, and 34mm. (1.34 inches) long. The other dimensions of the nozzles, however, were varied. The tubes were connected to the steam supply by a 2-inch pipe and care was taken

*Proceedings of the Engineers' Club of Philadelphia, July, 1891, from which the following abstract was prepared.

to secure dry steam. In order to determine the pressures within the nozzle seven small holes were drilled equal distances apart in the walls of the nozzle, commencing at the point where the curve of approach becomes tangent to the cylindrical barrel of the tube. Each of these apertures had gauge connections at the outer end, and the holes not in use were closed by plugs. (See Fig. 1.) A small searching tube was used for finding the internal pressure of the jet at points beyond the end of the nozzle. The tube was closed at one end and had holes drilled in one side near the end. The

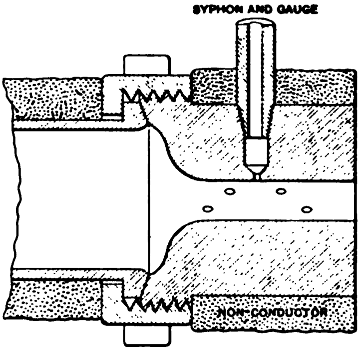


Fig. 1. Method of Measuring Pressures.

other end of the tube was connected with a gauge, and by placing the tube concentric with the axis of the nozzle and sliding it to different positions the pressures could be determined. Since the tube changed the relation of the areas of the different sections of the nozzle in which it was inserted, the nozzle was made of proportionately larger diameter to compensate for this, when the tube was used.

Mr. Kneass was the first in this country to make systematic investigation of steam discharge through nozzles. The questions of internal pressures, the relation between length and terminal velocity, and the taper and shape of nozzles were carefully gone into.

Explanation of Diagram.—The diagram, Fig. 2, shows a longitudinal cross section of each of the five nozzles tested. The vertical lines, *1, 2, 3, 4,* etc., cutting each nozzle pass through the points at which the pressures were measured. Below each nozzle section are plotted pressure curves passing through points corresponding

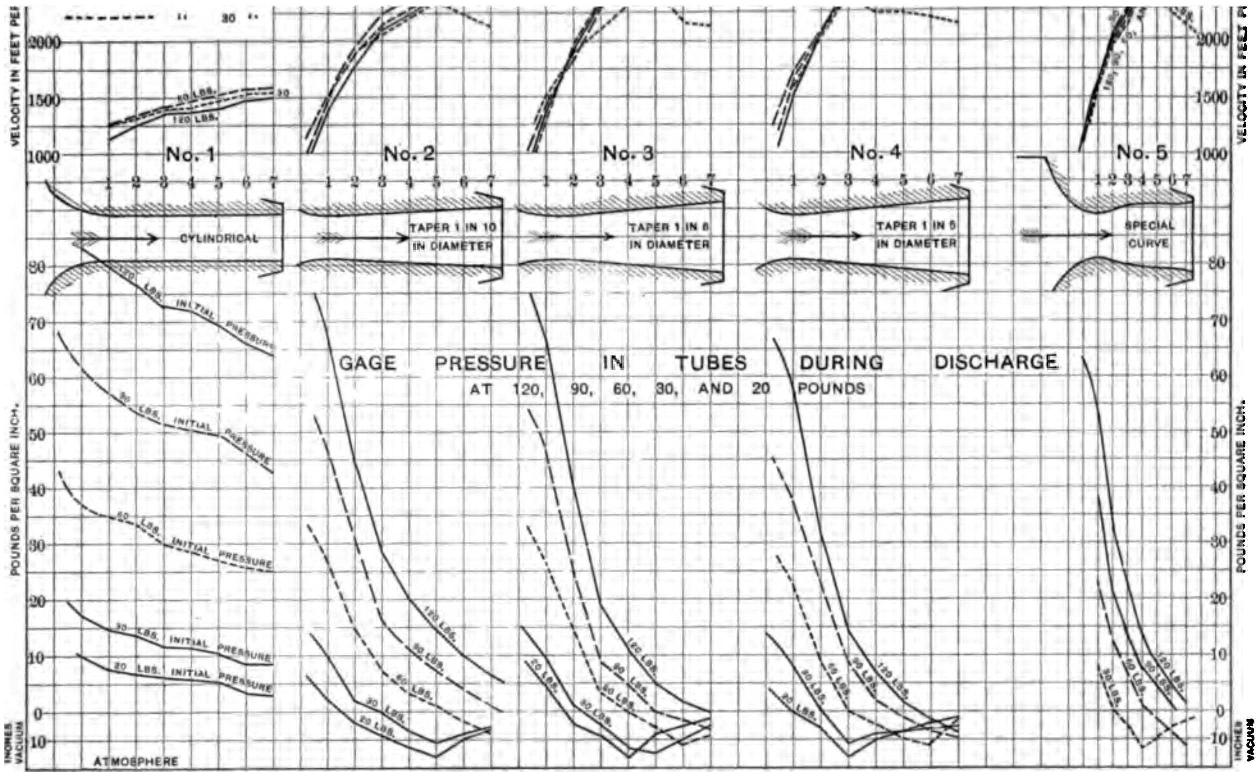


Fig. 2. Experiments by Strickland L. Kneass.

Tests with Nozzle No. 2.—In this nozzle, diverging with a straight taper of 1 in 10 in diameter, the throat pressure falls to about $6/10$ of the initial and is very nearly the same as the terminal pressure in the cylindrical tube. With high initial pressure the expansion line approaches the hyperbolic curve and discharges very close to the atmospheric line, while the 60-pound curve falls even into the vacuum lines and issues from the tube in a jet that contracts to a smaller diameter than that of the orifice. This phenomenon occurs with the 30-pound and 20-pound lines within the confines of the nozzle and the pressure at the mouth is greater than that at the fifth orifice. This shows that for these lower pressures there is a loss in efficiency in going beyond this point of maximum expansion, and that the discharging jet would have higher terminal velocity if the tube were made shorter. When the jet expands within the nozzle to a pressure lower than 15 pounds absolute, there will be a contraction of the jet after leaving the tube, due to the pressure of the atmosphere. These important facts were probably first noted in this series of experiments.

An experiment with the 1 in 10 tube, before and after reaming the discharge end with steeper tapers, showed that such change in the angle of divergence produced no effect upon the steam discharge of the inlet half of the tube, as the curves of velocity and pressure are identical for the inlet end of the tube under both conditions. The experiment has also been tried of gradually shortening the tube by cutting sections off the outlet end, and it was found that the pressures nearer the inlet were not affected by the alteration.

Design of Nozzle No. 5.—In all the straight nozzles there is loss of energy, due to fluid friction, owing to changes in velocity. Mr. Kneass reasoned that these changes in velocity should cause less loss and internal friction if the acceleration of the jet were made uniform. Nozzle No. 5 was designed to give constant acceleration. The pressure at the throat was assumed at its lowest theoretical value and the terminal velocity at the highest attainable in expanding from 120 pounds to the atmosphere. The tube was divided into seven parts and the acceleration calculated. Then, in order to divide the work evenly, the expansion was made uniform and from these data the area of the tube at each given section was

determined. As shown by the diagram, this tube gives superior results.

Deductions from the Velocity Lines.—It will be noticed that, notwithstanding the marked differences in initial pressure, ranging from 30 to 120 pounds, gauge, the velocity curves for the several nozzles, and particularly for nozzle 5, fall very close together. In the latter nozzle the velocities at any given point are practically the same for all the pressures used. Mr. Kneass finds by analyzing formulas of Rankine for the flow of steam that if the ratio of the throat to the initial pressure were the same under all conditions, there would be a lower velocity for 30 than for 120 pounds initial pressure at any given point in the nozzle. It will be seen from Table III., however, where this ratio is given for the different nozzles and for the different initial pressures, that its value decreases with the initial pressure. In other words, the throat pressure is a smaller percentage of the initial pressure when steam is being used at a low initial pressure than when used at a high initial pressure. This condition, apparently, affects the velocity at any point in the nozzle in a way partially to neutralize the effect of the initial pressure upon the velocity. This is shown both by the formulas and the tests. For example, in nozzle 5, taking the actual throat pressures as found by the tests, the velocities at point 4 for the different initial pressures are found by calculation to vary only by 4 per cent.

In Table IV. are the velocities of the steam for the different points in the several tubes as calculated from the experimental results. The variation of the velocity from a constant value for all the pressures in the table is small. Mr. Kneass states that for practical purposes the velocity may be considered constant and a simple formula derived for the weight of steam discharged through an orifice in a given time which is more convenient to apply than the thermodynamic equation. It is based on the assumption of constant velocity of discharge and is to be applied where the ratio of final to initial absolute pressures does not exceed 0.6.

Let W=weight in pounds discharged per second,
a=area of orifice in square inches,
d=weight of 1 cubic foot of steam in its initial condition.

Then, $W=6.19\ ad.$ (3)

Table III.

Ratio of Throat Pressure to Initial Pressure.

(Both in absolute pressures.)

Initial pressure, gauge lbs. sqr. in.	Shape of nozzle.					
	Cylindrical.	1 in 10.	1 in 6.	·1 in 6 and 1 in 10—mean.	1 in 5.	Special curve.
120	.708	.610	.606	.608	.539	.517
90	.689	.589	.582	.589	.498	.520
60	.665	.558	.558	.560	.501	.508
30	.664	.552	.541	.552	....	.507
20	.654	.524	.553	.520	.430	....
10	.676	.510	.545	.506?	.496	....

Table IV.

Steam Velocities in Nozzles.

Shape of tube.	Initial pressure (gauge).	Velocity in feet per second.						
		Sections of nozzle.						
		1	2	3	4	5	6	7
Cylindrical.	120	1195	1252	1349	1368	1400	1464	1500
	60	1261	1291	1416	1445	1513	1560	1589
Straight taper, 1 in 10.	120	1386	1799	2081	2250	2389	2495	2668
	90	1541	1946	2246	2378	2166	2601	2707
	60	1532	1882	2168	2272	2384	2459	2601
	30	1517	1926	2047	2235	2341	2235	2143
Straight taper, 1 in 6.	120	1400	1927	2328	2500	2669	2765	2855
	90	1483	1822	2312	2457	2664	2707	2867
	60	1310?	1960	2389	2476	2484	2712	2652
	30	1518	1927	2119	2380	2479	2158	2139
Straight taper, 1 in 5.	120	1504	2091	2466	2632	2801	2910	2975
	90	1710?	2062	2420	2526	2646	2768	2827
	60	1638	2071	2447	2483	2678	2755	2505
	30	1599	2042	2408	2216	2235	2206	2158
Special curve, uniform acceleration. Nozzle 5.	120	1662	2065	2272	2476	2657	2740	2890
	90	1681	2069	2267	2462	2563	2663	2823
	60	1696	2023	2216	2457	2592	2717	2881
	30	1626	1956	2187	2466	2331	2206	2148
Mean for nozzle 5 (continuous expansion).		1666	2028	2235	2465	2604	2708	2864

Example.—$a=1$; initial pressure=100 pounds absolute; $d=.2271$ (from steam table). Then, $W=6.19\times1\times.2271=1.405$ pounds per second. This is to be compared with the same example solved by equation (1), this chapter, according to Napier's rule.

Experiments on Steam Jets by Walter Rosenhain.*

Description of Apparatus.—In this series of experiments both cylindrical and diverging nozzles were used, and the velocity of discharge was measured by the ingenious plan of weighing the reaction of the jet upon the nozzle by means of a special apparatus. Knowing the reaction of the jet in pounds and the weight of steam discharged in a given time, the velocity could be accurately determined.†

In Fig. 3 is a diagram of the apparatus. A vertical tube, F, is joined to a steam pipe, D, and supports at its lower end a cylindrical chamber, H, which has a circular opening for the nozzle. The tube, F, is of bicycle tubing, which has sufficient flexibility to allow the cylindrical chamber to swing freely about the point of support at D. The reaction of the jet is measured by weights placed in the scale pan, S, carried by a cord passing over the pulley, P, and attached to the horizontal arm at K. A pointer indicates the movement of the arm and chamber, H, on the scale shown above the arm. Pulley P is a finely finished steel disk, with ball bearings. A gauge registers the pressure of the steam in H. In the tests the pressure of the steam supplied the apparatus was controlled, either by varying the boiler pressure or by throttling the steam after it left the boiler. After the observations of reaction were made, the discharge was measured in pounds per second under as nearly the same conditions as possible, the steam being conducted to a surface condenser and weighed.

The formula for the velocity of discharge is based on the old

*Proceedings of The Institution of Civil Engineers, London, Vol. CXL., 1900.

†A series of articles in London *Engineering*, 1872, upon "Experiments and Researches of the Efflux of Elastic Fluids," by Wilson, describes a similar method for measuring the velocity of a jet. Wilson's apparatus was elaborate and his tests were exhaustive, but the results are not of value for the present purpose. Another method has been used by Strickland L. Kneass, who arranged the nozzle to discharge against a delicately balanced parabolic target. The target turned the jet through an angle of 90 degrees and the pressure against the target should give a result equivalent to the reaction of a swinging nozzle.

ciple that action and reaction are equal,—the accelerating force ıe jet is equal to the reaction of the jet upon the nozzle and its nber, H,—and may be derived thus: Suppose a force, F, act- as a constant pull or push on a free body, to give the body a city of V feet per second at the end of one second. Then, since

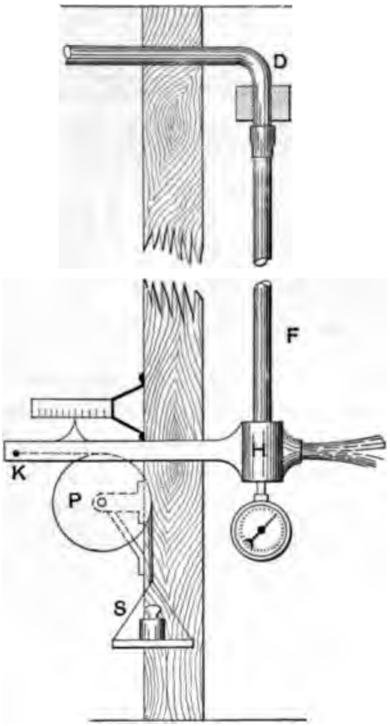


Fig. 3. Method of Measuring Velocity of Flow.

know that gravity, acting on the same body with a constant e of W pounds, equal to the weight of the body, would produce locity of 32.2 feet per second at the end of one second, we have,

$$F : W = V : g, \text{ or},$$

$$V = \frac{Fg}{W}. \qquad (4)$$

In applying this formula to the steam jet, we have:
V=velocity in feet per second,
F=reaction in pounds,
g=acceleration of gravity (=32.2),
W=pounds steam discharged per second.

TABLE V.

EXPERIMENTAL NOZZLES.

No.	Least Diameter.	Greatest Diameter.	Length.	Taper.
I.	0.1873	......	....	Thin plate
II.	0.1840	0.287	2.1	1 in 20
II. A	0.1866	0.1866	0.5	....
II. B	0.1849	0.287	1.6	1 in 20
III.	0.1883	0.308	2.16	1 in 12
III. A	0.1888	0.255	0.79	1 in 12
III. B	0.1888	0.241	0.64	1 in 12
IV.	0.1880	0.255	2.16	1 in 30

The Nozzles Used in the Tests are represented in Fig. 4, and their dimensions are tabulated in Table V. The nozzles were all made with a throat diameter as nearly as possible 3-16 inch in diameter, the exact dimensions being given in the table. This is the diameter of the nozzles of a De Laval 5-horse power turbine. The dimensions of De Laval nozzles of this size for several different pressures are given in Table VI., which shows the tapers to vary from about 1 in 17 to 1 in 27. Guided by this, the tapers of the experimental nozzles were made 1 in 12, 1 in 20, and 1 in 30, which gave wide enough latitude for the tests and yet kept reasonably close to proportions that had proved satisfactory in practice.

TABLE VI.

DE LAVAL NOZZLES FOR 5-H. P. TURBINE.

Pressure lbs. sq. in.	Least Diameter.	Length.	Taper.
	In 1.	In 1.	
136	0.157	1.57	1 in 17.4
105	0.163	1.57	1 in 21.4
100	0.197	1.57	1 in 19.0
60	0.230	1.57	1 in 29.0
58	0.256	1.57	1 in 26.6

Preliminary Experiments in Weight and Velocity of Flow.—As evident in Fig. 4, nozzles *II. A* and *II. B* are simply the two separate parts of nozzle *II.*, which has a well rounded inlet and a taper in the diverging part of 1 in 20. Preliminary experiments were run with nozzles *I.*, *II.*, *II. A,* and *II. B,* and the results are plotted in the diagrams in Figs. 5 and 6. Referring to the former,

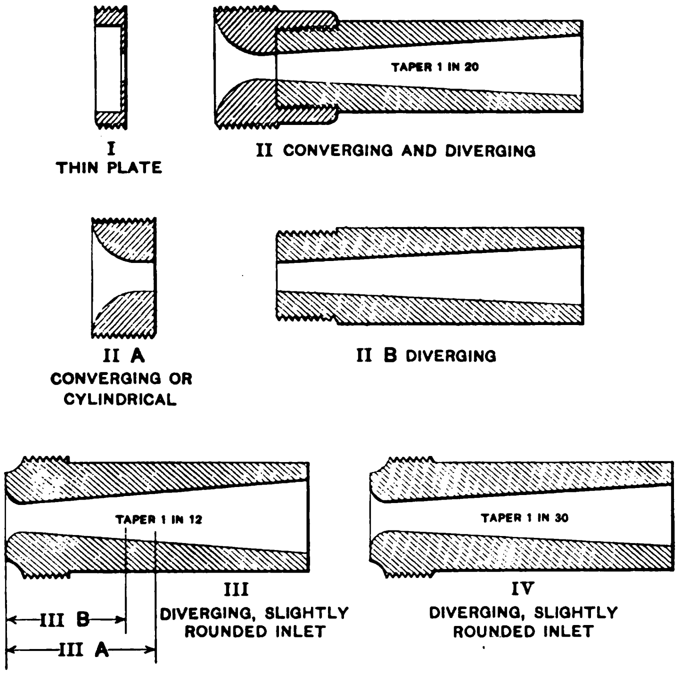


Fig. 4. Nozzles Used by Rosenhain.

which gives the weight discharged per second for different pressures, nozzle *II.*, having an easy inlet and an expanding outlet, gives the greatest discharge. Next comes the converging mouthpiece, *II. A,* which indicates that the inlet section of nozzle *II.* is more important than the outlet section in its effect upon the quantity of discharge. The position of *II. B* so far below *I.* seems to show that the sharp inlet is unsuited to passing a large quantity of steam through an expanding nozzle.

The velocity curves in Fig. 6, on the other hand, show that while

the quantity of steam passed by a nozzle depends very considerably on the shape of the inlet, the velocity on leaving the nozzle depends more on the shape of the outlet portion. This points to the conclusion that the density of the steam at the throat of the nozzle depends upon the shape of the inlet and that this density is greater with a well-rounded inlet than with a nozzle having a sharp inner edge.

This accounts for the most conspicuous feature of this set of velocity curves, viz., that up to a pressure of about 80 pounds per

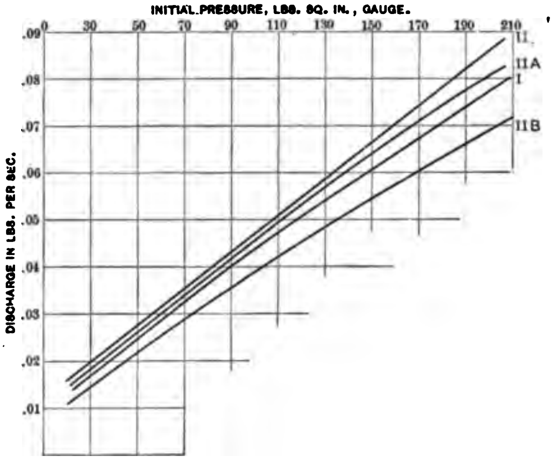


Fig. 5.

square inch the greatest velocity is attained by a jet from an orifice in a thin plate and that above 100 pounds per square inch, *II. B,* having a sharp inlet, gives a greater velocity than *II.,* which has a rounded inlet and the same outlet. Apparently the rounded inlet admits a greater weight of steam to the narrowest section than the nozzle can deal with efficiently.

The reason, of course, why the diverging nozzles give greater velocities at the higher pressures than the thin plate or the converging nozzle is that expansion is not complete in the two latter at the higher pressures and there is wasted energy.

Comparing the dimensions of nozzle *II. B* with the dimensions of the De Laval nozzles, it is found to lie midway between the ex-

tremes, and hence two other nozzles were designed, *III.* and *IV.*, having tapers, respectively, of 1 in 12 and 1 in 30, as previously stated. These, however, were made with inlet rounded slightly, with small radius, instead of with a large radius, as in nozzle *II.*; this in view of the fact that the preliminary tests had shown that when the inlet was considerably rounded the velocity of flow would not be as great.

Tests with Nozzle III.—In order to secure information upon the best ratio of outlet to throat diameters of a steam nozzle, tests

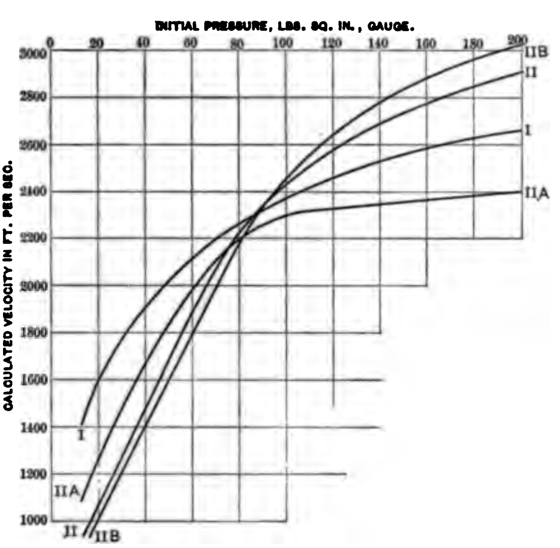


Fig. 6.

were run with nozzle *III.* at its original length of 2.16 inches and then it was cut down, first to .79 inches and finally to .64 inches long. Nozzle *III. B* discharged the greatest quantity of steam, nozzle *III.* slightly less and *III. A* the least. The discharge curves keep very close together, however, and it is not thought necessary to reproduce them herewith, as their mean values very nearly coincide with curve *II. A,* Fig. 5, for the converging nozzle. The velocity curves are given in Fig. 7, which shows nozzle *III. A* to be the most efficient. Comparing the curves in Fig 7, it seems probable that in nozzle *III. B* expansion was insufficient and that

in nozzle *III.* it was too great. It may be supposed that in the latter the steam expanded down to atmospheric pressure and then flowed through the remaining portion of the nozzle much as an incompressible fluid would do, the steam increasing in section at the expense of the velocity. (See velocity diagrams, also, in Fig. 2.)

Nozzle *III. A,* with 1 in 12 taper, is roughly comparable with nozzle *II. B,* which showed up the best of the 1 in 20 tapers, and from their velocity curves there appears to be a small difference in favor of the 1 in 12 taper.

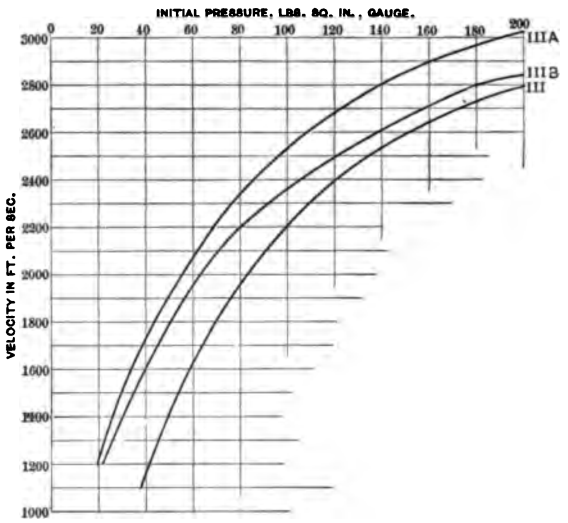


Fig. 7.

Tests with Nozzle IV.—It will be noted from Table V. that nozzle *IV.* is so designed as to be directly comparable with nozzles *III.* and *III. A.* It has the same length as nozzle *III.,* but a smaller outlet diameter, since its taper is 1 in 30 instead of 1 in 12. Its outlet diameter is the same, however, as of nozzle *III. A,* but its length, of course, is greater. The velocity curves for these three nozzles are plotted in Fig. 8. Nozzle *III.* gives the poorest results, as before, in diagram in Fig. 7, and *III. A* gives the best results, as before, while nozzle *IV.* falls between. The conclusion has already

been drawn that nozzle *III.* carried the expansion too far, which accounts for its position on the diagram; while as between *III. A* and *IV.*, both of which expand the steam to the same extent, it is very evident that *III. A*, which has the taper of 1 in 12, is much the superior.

After experimenting with nozzle *IV.*, Rosenhain measured the flow after the nozzle had been shortened by ⅜ inch at a time, until it was only .66 inch long, but the results have no special significance beyond what has been shown by the other tests.

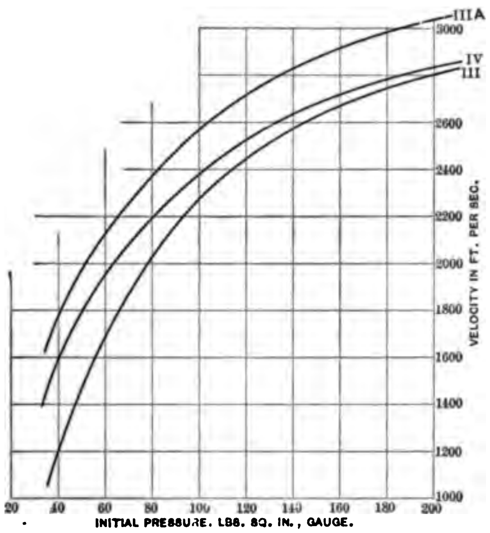


Fig. 8.

Conclusions from Rosenhain's Tests.—Rosenhain concludes from his experiments that the most efficient form of nozzle depends upon how great the pressure is. Up to 80 pounds, gauge pressure, an orifice in a thin plate is the most efficient form used in these experiments, but this does not imply that it is the most efficient form which can be used.

For high boiler pressures, an expanding nozzle with inner edge only slightly rounded should be used.

The taper should not be very different from 1 in 12, and the

proper ratio of greatest and least diameters is given, according to present results, in the following table:

Steam pressure in pounds per square inch......	80	100	140	160 to 200
Ratio of diameters..............................	1.26	1.26 to 1.33	1.36	1.36

Rateau's Experiments.*

Professor A. Rateau, of Paris, the inventor of the Rateau steam turbine, has made experiments upon the escape of steam through circular orifices. He experimented with the nozzles and the orifice shown in Fig. 9, the diameters of which are as follows:

Nozzle.	Millimeters.	Inches.
A	10.49	.412
B	15.19	.598
C	24.20	.954
D	20.12	.793

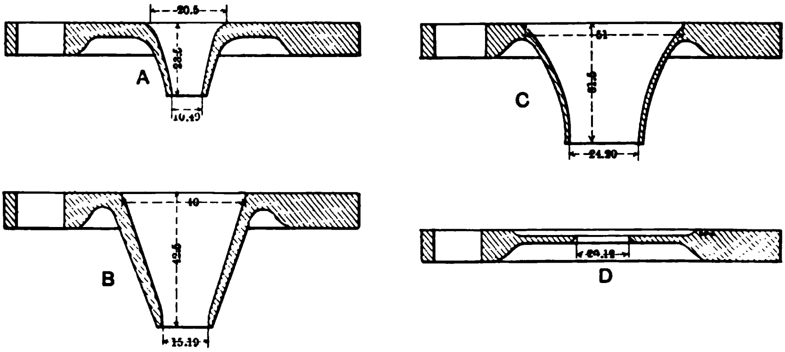


Fig. 9. Nozzles Used by Rateau.

Tests with Final Pressure Less than .58 Initial Pressure.—In this series a large number of tests were made, and a few data and results for each of the nozzles have been selected from the tables published by Rateau and grouped in Table VII. herewith. The original metric units and their English equivalents are given. In the case of the three converging nozzles the weight of steam discharged depends only on the initial pressure and is not affected by

*Paper by A. Rateau upon *"L' Ecoulement de la Vapeur D' Eau par Des Tuyers et des Orifices"* in the *"Annales des Mines,"* Paris, January, 1902. Since this abstract was made this paper has been translated and is now published by the D. Van Nostrand Company, New York.

Table VII.

Flow Through Converging Nozzles and Orifice in Flat Plate.

Nozzle.	Absolute pressures. Initial P. Kgs./cm².	Initial P. Lbs./inch².	Final p. Kgs./cm².	Final p. Lbs./inch².	Ratio $\frac{p}{P}$	Flow per second = I Grams/cm.²	Flow per second = I Lbs./inch²	Ratio of I to P. Metric.	Ratio of I to P. English.
A	9.16	130.30	.231	3.28	.025	133.34	1.89	14.52	.0145
	10.41	148.	.151	2.145	.015	149.96	2.13	14.40	.0144
	8.46	120.40	.131	1.85	.016	123.32	1.75	14.57	.0146
	10.14	153.40	.151	2.145	.014	150.12	2.26	14.31	.0148
	10.74	152.70	4.89	69.60	.455	152.96	2.17	14.24	.0142
	10.43	148.30	5.90	85.30	.571	149.43	2.12	14.32	.0143
B	9.26	131.85	.144	2.05	.016	132.43	1.88	14.31	.0143
	8.43	119.85	.144	2.05	.017	122.03	1.74	14.49	.0145
	6.68	95.	.144	2.05	.022	97.97	1.39	14.68	.0147
	4.48	63.70	.089	1.27	.020	67.01	.95	14.90	.0149
	5.27	74.9	.339	5.25	.070	76.52	1.08	14.52	.0145
	5.39	76.7	.249	3.54	.046	77.83	1.11	14.41	.0144
C	2.44	34.65	.154	2.19	.063	36.39	.517	14.92	.0149
	1.15	16.33	.127	1.81	.110	17.46	.248	15.19	.0152
	2.93	41.63	.171	2.43	.058	43.31	.616	14.78	.0148
	4.03	57.25	.235	3.34	.055	58.30	.829	14.46	.0145
	2.97	42.20	.931	13.70	.322	43.76	.621	14.73	.0147
	1.19	16.91	.644	9.16	.541	17.98	.255	15.11	.0151
D	4.04	57.50	.182	25.88	.450	51.92	.739	12.85	.0128
	3.86	54.90	.993	14.12	.257	49.45	.703	12.82	.0128
	2.57	36.56	.958	13.62	.373	32.85	.467	12.78	.0128
	2.36	33.58	1.38	19.58	.584	27.13	.385	11.50	.0115
	4.14	58.80	1.35	19.28	.321	51.27	.728	12.40	.0124
	3.84	54.59	1.88	26.80	.492	45.55	.647	11.88	.0119
	2.98	42.50	1.59	22.65	.534	35.18	.500	11.73	.0117

the final pressure. This is shown by the last two columns of the table, which give the discharge per kilogram, or pound, initial pressure, according as metric or English units are taken. These columns express the ratio,

$$\frac{I}{P} = \frac{\text{Wt. discharged per second per unit area}}{\text{Initial pressure per unit area}}$$

It will be noted that this ratio is very nearly a constant quantity except for the orifice, *D*, in the thin plate, showing that flow through the latter does not follow the same law.

Rateau plotted the results of these tests, using as ordinates the ratios I/P and as abscissas the discharge per second per unit area. He then plotted the theoretical discharge line, using a formula derived by the principles of thermodynamics, in order to compare results. The differences do not usually exceed two per cent. The actual discharge was slightly in excess of the theoretical, the mean deviation for nozzle A being .012 of the actual discharge; for nozzle B, .007, and for nozzle C, .003 of the actual. The difference in the sizes of the nozzles apparently had no marked effect on the results.

Formula for Weight Discharged.—From the foregoing it is evident that a single formula may be employed to calculate approximately the weight discharged for all pressures within the limits of the tests, provided the final pressure is not more than .58 of the initial. The following is proposed, the form of which is derived by theory, but the constants of which are taken from the tests:

$$I = P\ (15.26 - .96 \log P),$$

in which I is the flow in grammes per square centimeter per second, and P is the initial pressure in kilograms per square centimeter. This may be called the formula of maximum discharge, since it gives the greatest quantity that will flow through the nozzle for a given pressure.

In English units it reduces to

$$I = .001P\ [15.26 - .96\ (\log\ P + \log\ .0703)],$$

where I is the flow in pounds per square inch per second and P the initial pressure in pounds per square inch.

Example.—Given, $P = 6.68$, metric; 95, English.

In metric units,

$$\begin{aligned} I &= 6.68\ (15.26 - .96 \times .8248) \\ &= 6.68 \times 14.47 \\ &= 96.6 \text{ grams per square centimeter per second.} \end{aligned}$$

In English units,

$$\begin{aligned} I &= .001 \times 95\ (15.26 - .96 \times .825) \\ &= .095 \times 14.47 \\ &= 1.374 \text{ pounds per square inch per second.} \end{aligned}$$

Tests with Final Pressure More than .58 of Initial Pressure.—Under this condition the discharge depends upon the final pressure as well as the initial pressure and the results may be represented by taking the ratio of the measured discharge to the maximum calculated discharge I, obtained by the formulas above.

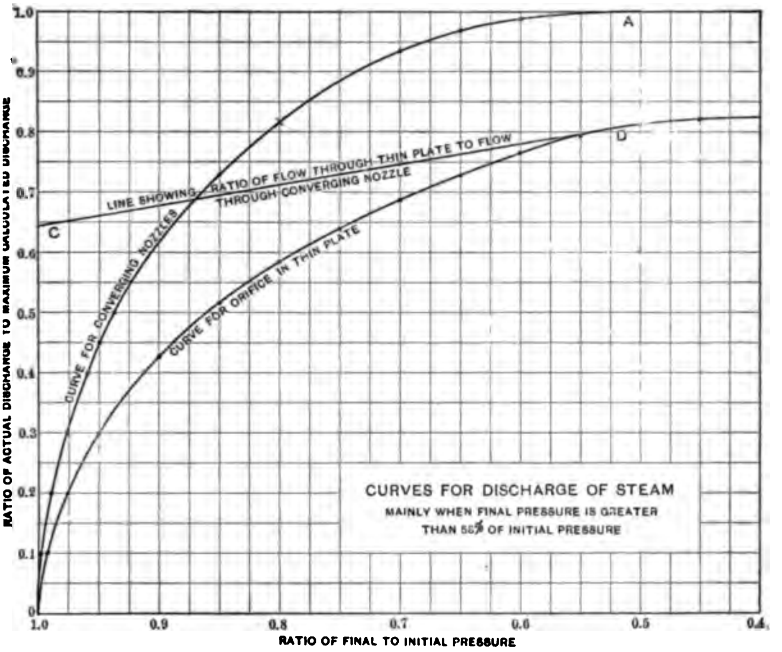


Fig. 10.

The diagram, Fig. 10, shows two curves, one for the converging nozzles and one for the thin plate, plotted on this basis. The ordinates represent the ratio:

$$\frac{\text{Measured discharge when } p \text{ is more than } .58\ P}{\text{Calculated discharge when } p \text{ is less than } .58\ P}$$

Where p is the final and P the initial absolute pressure. The abscissas give the ratio p/P.

Flow Through Orifice in Thin Plate.—Different laws govern this case. The discharge does not become the maximum for final pressure equal to or little less than .58 P, but it increases con-

American Society of Naval Engineers for May, 1904. Nozzles, orifices and passages of various forms were tested, the shapes shown in Fig. 11 being those best adapted to turbine work. The tests were conducted in groups, in each of which the initial pressure was kept constant and the final pressure varied to give the desired pressure differences. This, it should be said, is the correct way in which to obtain comparative results and it is unfortunate that more experimenters have not adopted the same plan. The weight and velocity of steam flowing from a nozzle depend more upon the initial pressure than upon the final pressure,

TABLE VIII.

COMPARISON OF FLOW THROUGH NOZZLES SHOWN IN FIG. 11.

Absolute initial pressure in atmospheres.	Absolute final pressure in atmospheres.	Flow in kilograms per hour—actual.				Ratio of flow.	
		Nozzle No. I.	Nozzle No. II.	Nozzle No. III.	Nozzle No. IV.	Nozzle I. to nozzle II.	Nozzle III. to nozzle IV.
1	2	3	4	5	6	7	8
9	8.8	28.1	39.9	32.5	49.7	.704	.653
9	8.5	43.2	55.9	57.2	88.8	.751	.646
9	8.	60.9	75.2	77.5	104.6	.811	.742
9	7.	....	94.2	90.7	105.4	....	.860
9	6.	90.0	106.1	91.3	105.6	.849	.867
9	1.	95.6	108.5	91.3	105.8	.880	.866
7	6.8	23.0	24.7	30.9	46.7	.930	.662
7	6.5	37.9	....	50.7	77.0	....	.656
7	6.	53.9	61.2	65.5	83.4	.880	.785
7	5.	68.3	78.6	72.0	82.8	.871	.879
7	4.	73.0	85.9	....	....	.851	....
7	1.	74.8	84.8	72.7	83.1	.882	.878
5	4.8	....	21.6	20.9	35.5	....	.590
5	4.5	31.9	....	39.3	59.9	....	.655
5	4.	42.5	49.4	50.5	61.0	.861	.829
5	3.	52.2	60.0	51.5	....	.876	....
5	1.	54.2	63.0	51.2	60.6	.864	.849
3	2.8	....	15.8	17.8	29.9	....	.595
3	2.5	24.6	29.2	29.2	36.8	.842	.794
3	2.	....	35.8	31.2	....	....	....
3	1.	32.7	38.3	32.1	37.0	.852	.868
2	1.8	....	12.2	14.1	21.4	....	.660
2	1.5	17.4	21.4	19.9	24.2	.815	.825
2	1.	22.7	24.9	20.6	24.1	.911	.855
2	.118	....	24.9	....	....	....	....

and tests in which the final pressure is kept constant and the initial pressure varied are not as satisfactory.

The two tables herewith were compiled from the results of about 300 tests recorded in the article mentioned. Table VIII is arranged to show the weights of steam discharged by each nozzle for different initial pressures and pressure differences. The first six columns are from the original data and the last two

TABLE IX.

[illegible] OF FLOW THROUGH CONVERGING AND DIVERGING NOZZLES.

[illegible]	[illegible]	Ratio of [illegible] pressure [illegible]	Flow in lb. per hour. Actual. Converging nozzle No. [illegible] [illegible] inches diameter at throat	Flow in lb. per hour. Actual. Diverging nozzle No. 4 [illegible] inches diameter at throat	Ratio of flow thro' diverging nozzle to flow thro' converging nozzle.	Flow in lb. per hour per sq. inch area at throat of diverging nozzle.
1	2	3	4	5	6	7
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	5500
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	6440
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]		.....	.....
[illegible]	[illegible]		[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
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columns were calculated by the author to facilitate comparisons.

Table IX contains certain data and results changed into English units and is arranged for the purpose of comparing the rate of flow from the straight and diverging nozzles, Nos. 2 and 4, with rounded inlets.

Investigations of Dr. C. E. Lucke.—In a paper presented before the American Society of Mechanical Engineers in January, 1905, Dr. Lucke of Columbia University records the results of experiments showing the pressures and temperatures of steam when flowing through a diverging nozzle such as is used in a De Laval

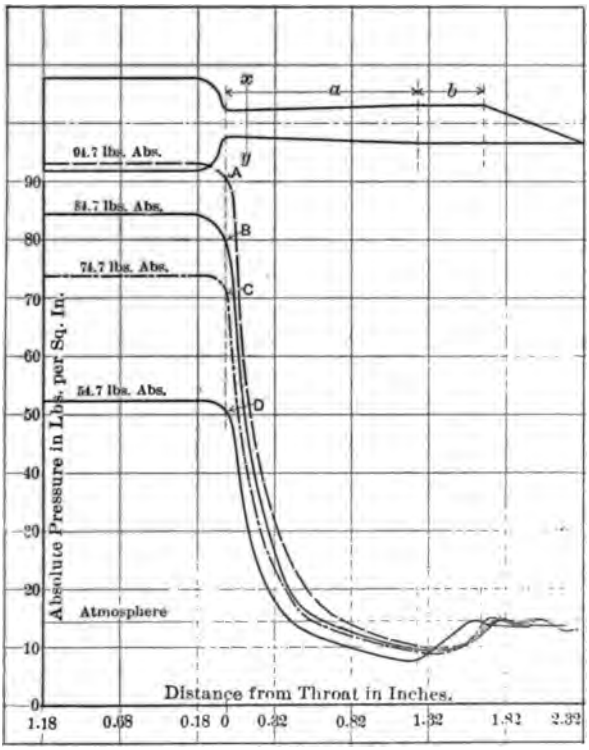

Fig. 12.

turbine. While no conclusions can be drawn from these experiments at present, owing to their incomplete state, the results indicate that the present method of converting the heat of steam

into kinetic energy in a nozzle may be less efficient than supposed, and that the commonly-accepted theory of the flow of steam through orifices does not conform to practice. In any case the tests show that further investigation is needed before we can assert that the passages of a steam turbine have been designed along the most successful lines possible.

In Fig. 13 is a sketch giving the dimensions of the nozzle. The pressures at the different points within the nozzle were determined in the usual manner by means of searching tubes. These were made in three different ways—one having an opening opposed to the current, one with the orifice opening in the direction of the current, and the third with the opening in the side of the tube. As a matter of course these three tubes gave widely different results, and it seems better, for the sake of comparison with other experiments, if for no other reason, to use the results obtained with the latter form of tube, since this is the type employed by other experimenters.

In the diagram, Fig. 12, are four curves showing the variation of pressure at different points within the tube, starting at initial pressures of 94.72, 84.72, 74.72, and 54.72 pounds absolute, respectively. The nozzle discharged against atmospheric pressure and was evidently designed for a greater pressure range of steam than was used. It will be noted that in each case the steam reaches atmospheric pressure midway between the inlet and outlet sections of the diverging portions of the nozzle, after which the effect is to cause what we have previously called "over-expansion" of the steam, the pressure dropping to 10 pounds absolute and then rising again to atmospheric pressure as the outlet of the nozzle is reached.

The theoretical critical point of 58 per cent of the initial pressure is not reached until a point about .12 inch from the throat is reached, represented by the line xy. The points in the different curves at which the pressure is 58 per cent of the initial pressure fall very nearly in the same vertical line. The actual throat pressures at A, B, C, and D are much higher than this.

Temperature Determinations.—In Fig. 13 is a diagram representing temperatures at different points in the nozzle for steam having an initial pressure of 84.72 pounds. The measured temper-

atures are represented by line *A*. The temperatures were determined by means of a thermo-couple of thin nickel copper wire which was stretched through the nozzle. It will be seen from the diagram and also from Table X. that the steam was superheated

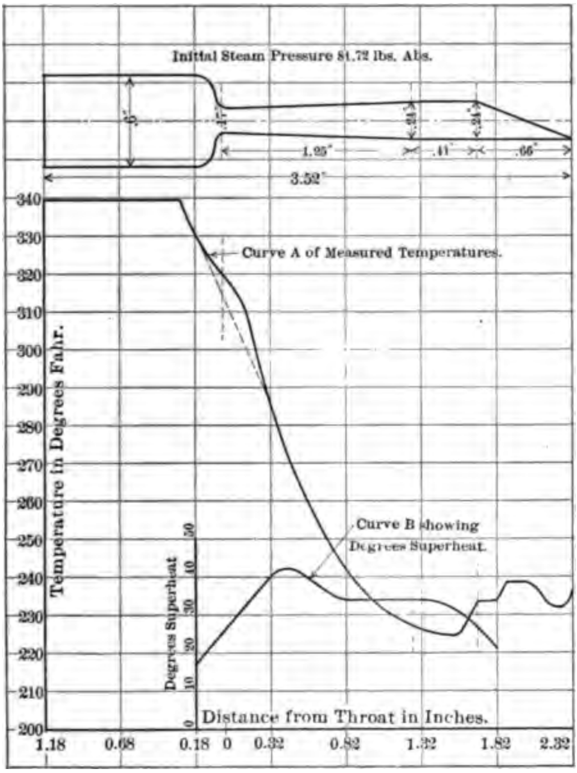

Fig. 13.

throughout the entire length of the nozzle. Because of throttling at the point of approach, the steam was superheated slightly at the start and remained so until it discharged. Line *B* shows the amount of superheating at different points, and indicates that the superheat increased to about the middle of the diverging part of the nozzle and then gradually decreased.

In the last column of Table X. is given the number of thermal units in the steam just before it enters the nozzle and at the point

Table X.

[illegible] in an Expanding Nozzle.

[illegible]	[illegible]	Corresponding Temperature [illegible] Steam	Measured Temperature.	Degrees Superheat.	B. T. U. in Steam=λ +.48×(t_s—t)
[illegible]	[illegible]	[illegible]	[illegible]	14.9[illegible]	11[illegible]6.4
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]	4[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]	54.5	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1157.3

[illegible] were calculated by taking the total heat [illegible] adding the amount of heat due to the [illegible] the number of degrees of super- [illegible] atmospheric pressure, which is .48. [illegible] heat present at the beginning and [illegible] of the steam converted into work, [illegible]

[illegible] at the start, however, and as [illegible] had been adiabatic, about 10 [illegible] been condensed and its heat [illegible] the heat represented by [illegible] times as much as was actually [illegible]

The [illegible] was so little heat [illegible]erted [illegible] steam superheat instead [illegible]ndense [illegible] experiments are to be [illegible]ted [illegible] be productive of much [illegible]le information.

CHAPTER XII

STEAM AND ITS PROPERTIES.

It is the purpose of this chapter to place before the reader the definitions, data, etc., which form the basis of steam calculations in steam turbine work. Such calculations pertain mainly to the flow of steam and to steam nozzle design, which are treated in the next chapter.

The following is a list of the symbols and their meaning which appear in this and the succeeding chapter.

NOTATION (ENGLISH SYSTEM ASSUMED).

p=absolute pressure, pounds per square inch.
P=absolute pressure, pounds per square foot.
t=temperature, degrees F.
T=absolute temperature.
J=mechanical equivalent of heat (778 foot-pounds).
q=heat of the liquid.
r=latent heat of vaporization.
λ (Greek letter lambda)=total heat of dry, saturated steam.
θ (Greek letter theta)=entropy of water.
ϕ (Greek letter phi)=entropy of steam.
c=specific heat at constant pressure.
v=specific volume (general expression).
s=specific volume of dry, saturated steam.
σ(Greek letter sigma)=specific volume of water=0.016 cubic foot.
V=velocity in feet per second.
g=acceleration due to gravity=32.2.
$\frac{V^2}{2g}$=kinetic energy of a jet in foot-pounds.
x=per cent dry steam in a mixture of steam and water.
A=area in square feet.
a=area in square inches.
W=weight in pounds.

Temperature.—The reference points of temperature are the melting point of ice and the boiling point of water at average atmospheric pressure. These points are 0° and 100° on the centigrade scale and 32° and 212° on the Fahrenheit scale. One hundred degrees C. correspond to 180 degrees F. Hence, to convert centigrade units to Fahrenheit units, multiply by $\frac{9}{5}$ and add 32.

To convert Fahrenheit to centigrade, subtract 32 and multiply by $\frac{5}{9}$.

Example.—140° C.$=140\times\frac{9}{5}+32=284$° F. Again, 5° F. (or 27° below freezing)$=(5-32)\times\frac{5}{9}=-27\times\frac{5}{9}=-15$° C., or 15° below zero, C.

Absolute Temperature.—The absolute zero of temperature, at which heat is supposed to be entirely absent, is theoretically 492.7 degrees F. below the freezing point of water. The demonstration of this given in treatises upon heat is based upon the law of the heat expansion of gases, under the assumption that the law holds at the extremely low temperatures. This assumption is in error, but to what extent is not known. Letting $T=$ absolute temperature and $t=$temperature on the ordinary scale,

$$T=t+460.7 \text{ Fahrenheit, and}$$
$$T=t+273.7 \text{ centigrade.}$$

In steam calculations absolute temperatures are not to be used unless it is so stated. The temperatures of the steam table are the ordinary temperatures.

Pressure.—The average atmospheric pressure is taken in this country to be 14.7 pounds per square inch. On the continent of Europe pressures are usually measured in atmospheres, and one atmosphere is taken to be equal to a pressure of one kilogram per square centimeter, which is equivalent to 14.22 pounds per square inch, instead of 14.7 pounds.

Gauge pressure is the pressure denoted by a steam gauge and is measured above the pressure of the atmosphere; that is, it does not include the pressure of the atmosphere.

Absolute pressure is equal to gauge pressure plus atmospheric pressure, usually taken at 14.7 pounds per square inch. The exact absolute pressure can be determined only by the use of the barometer, which gives the pressure of the atmosphere in inches of mercury. One cubic inch of mercury weighs 0.49 pound at 60 degrees F. and 30 inches of mercury are therefore equivalent to 14.7 pounds pressure at this temperature.

In steam calculations absolute pressures are to be used instead of gauge pressures, unless otherwise stated.

Vacuum is measured in inches of mercury or millimeters of

mercury. The vacuum gauge used in commercial work shows the height in inches of a column of mercury that the pressure of the atmosphere will support against the pressure that is being measured.

Example.—If a vacuum gauge attached to a condenser reads 26 inches, and the barometer stands at 30 inches, what is the absolute pressure in the condenser? Since the barometer stands at 30 inches, the atmospheric pressure is $30 \times 0.49 = 14.7$ pounds per square inch. $30 - 26 = 4$; and the pressure in condenser$=$ $\frac{4}{30} \times 14.7 = 1.96$ pound per square inch.

Heat Unit, or Thermal Unit.—In the English system heat is measured in British thermal units (B. T. U.). A British thermal unit is the amount of heat necessary to raise one pound of water from 62 degrees F. to 63 degrees F. In the French system the calorie is the unit, equal to 3.968 B. T. U.

Mechanical Equivalent of Heat.—This is the number of units of mechanical work to which one unit of heat is equivalent.

1 B. T. U.=778 foot-pounds.
1 calorie=426.9 meter-kilograms.

Specific Heat is the number of thermal units required to raise unit weight of a substance one degree temperature. The specific heat of water is unity at temperatures ranging from 59 to 68 and from 104 to 113 degrees F., and approximately unity at other temperatures. The specific heat of nearly all other substances is less than one.

The foregoing is properly the definition for true specific heat. Sometimes, when a substance is raised through several degrees temperature, it is necessary to find the mean specific heat between these limits of temperature. This is the average number of thermal units per degree required to raise a unit weight of a substance from one temperature to any other given temperature.

Specific Volume.—In problems relating to the expansion of gases, it is convenient to deal with unit weights of the substance and to consider the volume occupied by unit weight. This is called specific volume. In the English system it is the cubic feet occupied by one pound and in the metric system the cubic meters occupied by one kilogram.

Specific Pressure.—When specific volumes enter into an example we have to deal with cubic feet instead of cubic inches and with cubic meters instead of cubic centimeters. Pressures used in carrying through the calculation must therefore be expressed in pounds per square foot or in kilograms per square meter, according to which system is being used. These are called specific pressures. Errors frequently creep into calculations by failure to note when specific pressures should be used.

Saturated Steam is steam generated in contact with water and the temperature of which always corresponds with the pressure.

Superheated Steam is steam heated to a temperature higher than that corresponding to the pressure, as in saturated steam. The superheating is produced by applying heat directly to the steam itself, instead of to the water from which it is generated, and if the superheating is to be carried far, the steam must be in a separate chamber, and not in contact with water.

Steam Tables.—Tables of the properties of saturated steam, used to facilitate steam calculations, contain columns of figures giving certain important properties of steam. The following are the most important headings for steam turbine calculations. Columns of the table giving heat units refer to the number of heat units in one pound of water or steam, as the case may be.

1. Absolute pressure, pounds per square inch.
2. Temperature of the boiling point corresponding to pressures of column 1.
3. Heat of the liquid, from 32° F.
4. Latent heat of vaporization.
5. Total heat in the steam and water, from 32° F.
6. Entropy of water.
7. Entropy of steam.
8. Specific volume—cubic feet per pound.
9. Density—weight of a cubic foot in pounds.

Of these, numbers 6 and 7 will be explained later, while numbers 3, 4, and 5 can be best illustrated by considering the several steps involved in the generation of steam.

The Generation of Steam.—In the operation of a steam boiler the pressure is so nearly constant that it may be assumed to be so during the evaporation of each individual pound of water. When

steam is generated under constant pressure, the process may be divided into several different steps, as follows:—

First Step.—Heating the water from 32° F., to the temperature of vaporization (equal to the temperature of the steam in the boiler). The number of heat units required is called the "heat of the liquid," represented by q.

Second Step.—Changing the water into steam, during which process the temperature does not change. The heat goes first to break up and separate the particles of water instead of to raise the temperature, as in the first step; and second, to increase the volume from that occupied by the water to that occupied by the steam. The number of heat units required for the second step is called the "latent heat of vaporization," or "latent heat," simply, represented by r. The total heat required to generate steam, including heating the water, is called the "total heat of the steam," and comprises the total heat energy of the steam. (Represented by λ.)

Third Step.—If superheated steam were produced, there would be a third step consisting in raising the temperature from that of saturated to that of the superheated steam.

Heat Required to Raise Temperature of Water.—Since the specific heat of water is approximately 1 at all ordinary temperatures, the number of heat units required to raise the temperature of one pound of water from t_1 to t_2 can be approximately calculated by subtracting t_1 from t_2, or

$$\text{Heat required} = t_2 - t_1. \qquad (1)$$

Also the heat q of the liquid can be found approximately by making $t_1 = 32$ degrees; thus,

$$q = t - 32 \qquad (2)$$

where t is the temperature of the water.

Heat Contained in Wet Steam.—Saturated steam condenses rapidly when its heat is converted into mechanical work, or when it comes into contact with a colder body, and in steam calculations we have generally to deal with a mixture of steam and water instead of with dry steam.

The quantity of steam present in a mixture of steam and water is expressed as a certain number of hundredths, x, of the total

weight of the mixture. One pound of the mixture contains a portion x of steam and $(1-x)$ of water. The total heat of the mixture is equal to the heat required to raise one pound of water to the given pressure, plus the heat required to evaporate the part x of the water into steam; or, $q+xr$. Hence total heat of one pound of wet steam $= xr+q$. (3)

Specific Volume of Wet Steam.—

Let s=specific volume of dry steam.

v=specific volume of wet steam.

σ=specific volume of water=1/62.4=0.016, since the weight of one cubic foot of water is 62.4 pounds.

Then, $$v=xs+(1-x)\sigma. \tag{4}$$

The last term, $(1-x)\sigma$, is so small that it can usually be omitted. At high pressures, say at 200 pounds, specific volumes have low values and the effect of the term $(1-x)\sigma$ is proportionately great. At 200 pounds pressure, however, its omission would cause an error of less than a tenth of one per cent for each 10 per cent of moisture present, which is less than probable errors in the steam table. We may, therefore, use for specific volume of wet steam, $$v=xs. \tag{5}$$

Total heat of Superheated Steam.—To determine this, we have,

$$\text{Total heat}=\lambda+c\ (t_s-t), \tag{6}$$

where

λ=total heat of dry saturated steam at the given pressure.

t_s=temperature of the superheated steam.

t=temperature of saturated steam at the given pressure.

c=specific heat of superheated steam at constant pressure. Values of this will shortly be discussed.

Example.—Find the total heat of superheated steam at 450 degrees F. and 100 pounds absolute pressure, assuming c=0.55. From the steam tables, λ =1,181.9 and t=327.3. Hence,

$$\text{Total heat}=1{,}181.9+0.55\ (450-327.3).$$
$$=1{,}181.9+67.5=1{,}249.4.$$

Adiabatic Expansion.—When steam expands without receiving or giving up heat it is said to expand adiabatically. Steam flowing through a correctly proportioned nozzle flows adiabatically, or

nearly so, because its passage is so rapid that little or no heat can be transmitted to it from any external source, nor can much heat be lost through radiation or otherwise.

Temperature-Entropy Diagram.

The graphical method of representing mechanical work is by means of the pressure-volume diagram, like the indicator card of a steam engine. In laying out such a diagram we use two co-

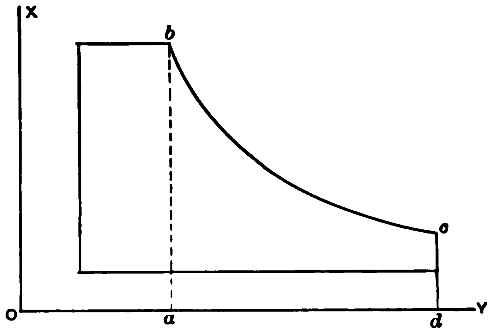

Fig. 1. Pressure-Volume Diagram.

ordinates, OX and OY, Fig. 1, drawn through point O, the zero of volume and pressure. Points on the diagram are determined by measuring volumes from OX and pressures from OY. The area of the diagram represents mechanical work.

Heat Diagram.—Similarly, heat energy may be represented by the area of a diagram constructed with vertical ordinates of absolute temperature and horizontal dimensions obtained by dividing the number of heat units added or subtracted during any change by the absolute temperature during that change. The horizontal distance of any point from the vertical axis OX of the heat diagram is called its Entropy, just as this distance on the work diagram represents volume. This term "entropy" gives the name *Temperature-Entropy Diagram* to the heat diagram.

The Analogy between the Work Diagram and the Heat Diagram may be further explained by selecting some one part of the diagram, Fig. 1, as the expansion line *bc,* which is reproduced in Fig. 2. The area *abcd* under this curve represents the work done

during the change brought about by the expansion from b to c. The mean pressure during this change is h. The volume at the start is represented by distance Oa; the volume at the end by Od; and the change in volume due to the expansion by ad.*

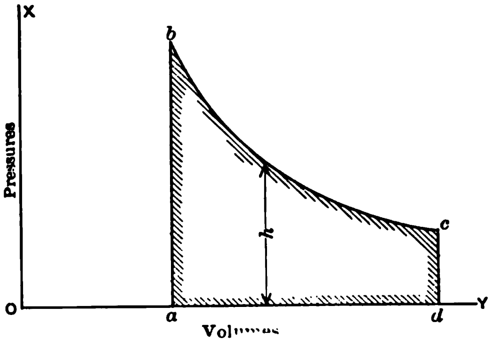

Fig. 2.

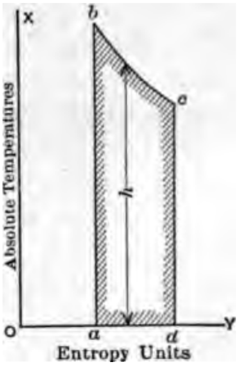

Fig. 3.

Now the area $abcd$=product of width by mean height, or ad by h.

Hence, for the pressure-volume diagram:—

$$\begin{pmatrix}\text{Work done in foot-pounds}\\ \text{during any change}\end{pmatrix} = \begin{pmatrix}\text{mean pressure}\\ \text{during change}\\ \text{in lb. per sq. ft.}\end{pmatrix} \times \begin{pmatrix}\text{change of}\\ \text{volume in}\\ \text{cubic feet.}\end{pmatrix}$$

In Fig. 3 is the corresponding temperature-entropy diagram, in which bc is the expansion curve showing a change, both in absolute temperature and entropy. The area $abcd$ under this curve represents the heat units given up by the working fluid during the expansion. The mean absolute temperature during the expansion is h. The entropy at the start is shown by the distance Oa; the entropy at the end by Od; and the change in entropy, due to the expansion, by ad. It is to be noted that the entropy is measured from the ordinate OX and that ad is not the entropy of point d, but that it is the *change* in entropy during the change in the condition of the substance.

*In the English system the work is in foot-pounds, the pressures are in pounds per square foot and the volumes in cubic feet. This is really what is represented by the indicator diagram, although the pressures are always measured in pounds per square inch instead of pounds per square foot. This is balanced, however, by taking the area of the piston in square inches instead of square feet, so the final result is the same.

Now, as before, area $abcd$=product of width by mean height, or $ad \times h$, and we therefore have the following statement for the heat diagram, analogous to the above statement for the work diagram:—

$$\left(\begin{array}{c}\text{Number of heat units added or}\\ \text{subtracted during any change}\\ \text{per pound of the working fluid}\end{array}\right) = \left(\begin{array}{c}\text{mean absolute}\\ \text{temperature}\\ \text{during change}\end{array}\right) \times \left(\begin{array}{c}\text{change of}\\ \text{entropy.}\end{array}\right)$$

From this we have,

$$\text{Change of Entropy} = \frac{\text{Heat units added or subtracted during change}}{\text{Mean absolute temperature during change.}}$$

This we will take for our definition of entropy.

Entropy of Water.—When water is heated its temperature and entropy both vary, which makes it necessary to use the calculus to obtain the equation for change of entropy.

Assume heat to be added to one pound of water, raising the temperature a small amount, dT.

If c is the specific heat of water, the heat added will be cdT.

Let $d\theta$ be the change in entropy corresponding to the change dT in temperature.

Now if these quantities dT and $d\theta$ be taken indefinitely small, the temperature during the change may be assumed constant and equal to T. Hence, by the definition of entropy,

$$d\theta = \frac{cdT}{T} \tag{7}$$

Let us assume the water to be heated from the temperature T_1 to temperature T_2, and the entropy to increase from θ_1 to θ_2. Then, by integrating between the limits T_1 and T_2, according to the principles of the calculus, we get for the change of entropy,

$$\theta_2 - \theta_1 = c \log_e \frac{T_2}{T_1} \tag{8}$$

Entropy is reckoned from 32 degrees F., or 492.7 degrees absolute, the same as the other heat properties of the steam table, and if T_1 be taken equal to 492.7, θ_1 becomes zero and we have for the entropy of water,

$$\theta = c \log_e \frac{T}{492.7} \tag{9}$$

$$= 2.3\, c \log \frac{T}{492.7} \tag{10}$$

It answers practical requirements to take the specific heat, c, in the above formulas equal to 1.

Entropy of Saturated Steam.—The total heat of steam is considered made up of two parts, the heat q required to raise the temperature of the water from the freezing point to the temperature of vaporization, and the latent heat r required to convert the water into steam. In like manner the entropy of steam consists of two parts, the entropy of water at the temperature of vaporization and the change, or increase of entropy that occurs when the water is changed into steam. The latter is sometimes called the ***entropy of vaporization,*** but more properly it is the ***change*** of entropy due to vaporization.

During vaporization the temperature remains constant and the change of entropy is easily calculated by dividing the latent heat r by the absolute temperature T, or,

$$\text{Change of entropy} = \frac{r}{T}$$

From this we have, for the entropy of steam, introducing x to make it general for either wet or dry steam (see formula 3),

$$\phi = \theta + \frac{xr}{T} \tag{12}$$

$$= 2.3\, c \log \frac{T}{492.7} + \frac{xr}{T} \tag{13}$$

Example.—To find the entropy of saturated steam at 100 pounds absolute pressure, we have, from the steam tables, $p = 100$; $r = 884$; $t = 327.58$, whence $T = 460.7 + 327.6 = 788.3$. Assuming steam to be dry,

$$\theta = 2.3 \log \frac{788.3}{492.7}$$

$$= 2.3 \times (2.89669 - 2.69258) = 0.47$$

Hence, $$\phi = 0.47 + \frac{884}{788.3} = 1.12$$

Entropy of Superheated Steam.—The entropy of superheated steam may be found by adding to the entropy of saturated steam the change in entropy due to superheating, which is expressed by the equation,

$$\phi_s - \phi = 2.3\ c \log \frac{T_s}{T} \qquad (14)$$

where $\phi_s - \phi$ is the change in entropy, c is the specific heat of superheated steam, and T_s and T are the temperatures of superheated and saturated steam, respectively, at the given pressure.

Example.—If the steam in the last example were superheated 250 degrees, what would be its entropy, assuming its specific heat to be 0.6? Here $T=788.3$ and $T_s=788.3+250=1{,}038.3$. Hence,

$$\phi_s - \phi = 2.3 \times 0.6 \times (3.01632 - 2.89669)$$
$$= 2.3 \times 0.6 \times 0.1196 = 0.165$$

and $\phi_s = 1.12 + 0.165 = 1.285$

Temperature-Entropy Diagram for Water and Steam.—In Fig. 4 the various heat changes for water and steam are shown graphically. Absolute temperatures are laid off on OX and values for entropy on OY. The different steps in the process of plotting the diagram are as follows:—

(1) Assume one pound of water at 100 degrees F. to be heated until its temperature reaches 350 degrees F. Its entropy increases and the change in temperature and entropy is represented by the water line ab. This curve starts on the ordinate OX, at the freezing point, and other points on the curve are calculated by the aid of equation (10); or they may be plotted from values of "entropy of the liquid" given in steam tables. If θ_1 is the entropy at point a and θ_2 at point b, then $\theta_2 - \theta_1$ is the change in entropy while the temperature increases from 100 to 350 degrees. The heat added during the change is represented by area a_1abb_1.

(2) If the water, at 350 degrees, is vaporized, the temperature will remain constant and the line bc will represent the change in entropy. At point c the vaporization is complete and its distance from OX is the entropy of steam at 350 degrees temperature, calculated by equation (12). The heat required for this change is shown by the area b_1bcc_1.

(3) In the flow of steam through nozzles, it is assumed that the expansion is adiabatic, no heat being added or subtracted. This condition can be represented only by the vertical line *cd*, which has no area under it to indicate heat added or subtracted.

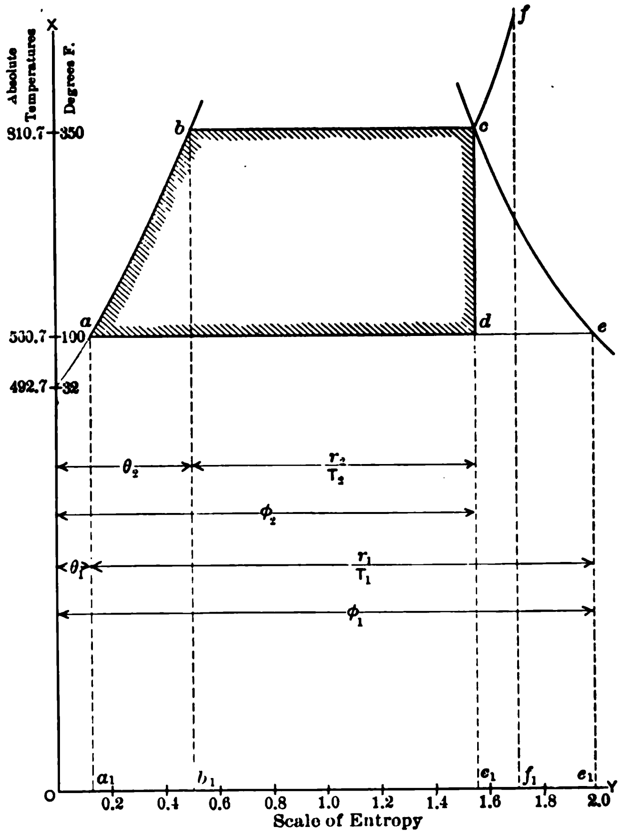

Fig. 4. Temperature-Entropy Diagram.

This line shows that while the temperature drops the entropy remains constant.

(4) To complete the cycle, assume the steam remaining at the end of adiabatic expansion to condense at constant temperature, as shown by the line *da*. The heat given up during condensation is represented by the area a_1adc_1. Deducting this area from areas $a_1abb_1+b_1bcc_1$, which show the heat applied, leaves

the net area *abcd,* which represents the available heat energy for doing work on this cycle.

Line for Dry Saturated Steam.—If, instead of adiabatic expansion, there were heat enough added to the steam while the temperature was dropping from 350 to 100 degrees to maintain the steam in a dry and saturated state, the conditions would be represented by the steam line *ce.* The heat required to maintain this condition is represented by c_1cee_1. The various points on the steam line are calculated by adding the entropy due to vaporization to the entropy of water for the corresponding temperatures. In other words, line *ce* shows the entropy of dry, saturated steam for the temperatures within its limits.

Line for Superheated Steam.—If at point *c* heat were added to superheat the steam the change would be represented by a curve like *cf,* plotted by the aid of equation (14). The heat required to superheat is shown by area c_1cff_1.

Conditions with Moisture Present.—In Fig. 5 *ab* and *ce* are the

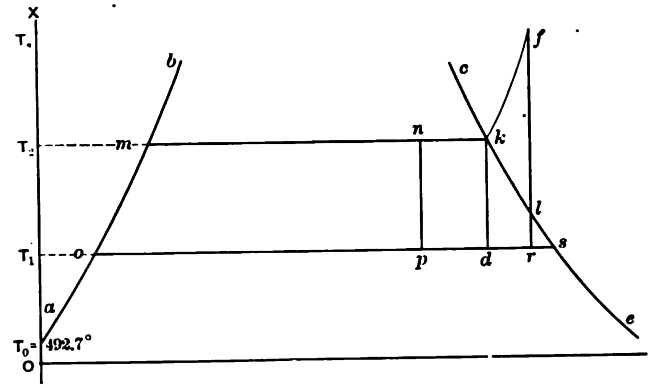

Fig. 5.

water and steam lines, respectively, of a temperature-entropy diagram. Take the case of one pound of water at temperature T_2, as indicated on the diagram. Its entropy is equal to the distance of point *m* to the right of line *OX.* If this water is all evaporated into steam, the conditions will be represented by point *k* on steam line *ce.* If, however, only a part of the water be evaporated into

steam, say 80 per cent, then the change of entropy due to vaporization will be

$$\frac{xr}{T}=\frac{0.8r}{T}$$

and the conditions will be represented by point n, which is 80 per cent of the distance from m to k. By the principles of percentage, therefore, the ratio

$$\frac{mn}{mk}$$

will give the percentage, x, of dry steam present.

The chief value of the temperature-entropy diagram for steam turbine work lies in the fact that it may be made to show the quantity of moisture present in steam by the method just indicated.

Thus, in the above case, suppose the steam to expand adiabatically to temperature T_1. By drawing the adiabatic line np, we find the percentage of dry steam present to be

$$\frac{op}{os}.$$

If dry, saturated steam expands adiabatically from T_2 to T_1, we find, by drawing adiabatic line kd that there is

$$\frac{od}{os}$$

per cent of dry steam present.

If superheated steam expands adiabatically from temperature T_s to T_1, draw the adiabatic line fr. At point l, where it intersects the steam line, the steam loses its superheat and becomes saturated, while at temperature T_1 it is only

$$\frac{or}{os}$$

per cent dry. If it were desired to know how far to continue superheating to secure dry, saturated steam at the end of expansion, the superheat curve kf must be extended until point f comes vertically over point s, which would show the desired temperature T_s.

Characteristic Equations for Adiabatic Expansion.

During adiabatic expansion heat is neither added nor abstracted and, as seen above, the change is represented by a vertical line on the temperature-entropy diagram. The entropy, therefore, remains constant during the adiabatic expansion, and by expressing this relation in the form of an equation the percentage of dry steam present at the end of adiabatic expansion may be easily calculated. This relation is expressed in the three equations which follow, in which the letters with subscript 1 refer to the higher pressure and those with subscript 2 to the lower pressure.

(1) For saturated steam,

$$\frac{x_1 r_1}{T_1} + \theta_1 = \frac{x_2 r_2}{T_2} + \theta_2 \tag{15}$$

(2) Steam superheated sufficiently to remain superheated throughout expansion,

$$\frac{x_1 r_1}{T_1} + \theta_1 + 2.3\, c \log \frac{T_{s1}}{T_1} = \frac{x_2 r_2}{T_2} + \theta_2 + 2.3\, c \log \frac{T_{s2}}{T_2} \tag{16}$$

(3) Steam superheated at the start but saturated at the end of expansion,

$$\frac{x_1 r_1}{T_1} + \theta_1 + 2.3\, c \log \frac{T_s}{T_1} = \frac{x_2 r_2}{T_2} + \theta_2 \tag{17}$$

The application of these formulas will be shown in the following chapter.

The Specific Heat of Superheated Steam.

Regnault's Result.—Until recently the value universally adopted for the specific heat of superheated steam at constant pressure has been 0.48, derived in 1840 from the results of three series of experiments by Regnault. In steam calorimeter work, where the temperatures came within the limits of Regnault's experiments, the value of 0.48 is practically correct, but with the high pressures and temperatures prevailing in power plants using superheated steam a higher value should be taken.

The Importance of a Correct Value.—In tests upon super-

heaters, or upon turbines and engines using superheated steam, the weight of the steam used does not afford a fair basis for estimating the gain or loss from superheating, because weight alone gives no indication of the amount of heat in superheated steam of a given pressure. If the specific heat of superheated steam were accurately known, however, this would give us the means of calculating the number of heat units in the steam and the efficiency of the apparatus could be determined on this basis, by the method explained under "The Thermal Unit Basis of Performance" in Chapter IX. The author has made calculations upon tests of a De Laval turbine, run first with saturated and then with superheated steam. By first taking the rate of water consumption in pounds as the basis for calculating the gain from superheating, the gain was found to be 8.1 per cent. Then, by taking as the basis the heat units in the steam, the gain was found to be as follows: Assuming specific heat=0.48, gain=4.8 per cent; specific heat=0.6, gain=4 per cent; specific heat=0.8, gain=2.7 per cent. These results show that the gain from superheating on the basis of heat units utilized is much less than when on the basis of pounds of water per horse-power per hour; and that the higher the value assumed for specific heat the less the gain is found to be. This illustration shows the importance of a correct value for the specific heat of superheated steam in calculating efficiencies.

*Results of Tests to Determine Specific Heat.**—Many experimenters have attempted to derive values for the specific heat of superheated steam at constant pressure for other pressures and temperatures than covered by the tests of Regnault. Among the more important work in this connection is that of Grindley in England, of Greissmann, Lorenz, and Messrs. Knoblauch, Linde and Klebe in Germany, and of Carpenter, Jones, Thomas, Burgoon, and engineers of the General Electric Company in America. The results of the various experimenters are more or less contradictory and it is not yet definitely settled how the specific heat varies in relation to pressure and temperature changes.

*The reader who wishes to investigate the subject of specific heat of superheated steam is referred to the *Journal of the Worcester Polytechnic Institute*, November, 1904, containing an article by Prof. Sidney A. Reeve; to *Power* for August, 1904, containing an article by Chas. A. Orrok; to the Stevens Institute *Indicator* for October, 1905, containing an article by Prof. J. E. Denton; and to the paper upon the "Steam Plant of the White Motor Car," by Carpenter, in the proceedings of the A. S. M. E., December, 1906.

Greissmann's and Grindley's Results.—An early review of the work of the different experimenters was contributed by Mr. George A. Orrok, chief draftsman of the New York Edison Company, to *Power* for August, 1904. He plotted the various values and found those of Greissmann to be the most consistent. Taking these as a basis Mr. Orrok deduces the following formula to represent them:

$$c = 0.00222\ t_s - 0.116 \tag{18}$$

in which t_s = temperature of the superheated steam.

The above formula gives the instantaneous or true specific heat at any temperature. The *mean* value of the specific heat between the points of saturation and any degree of superheat can be found by the formula

$$c = 0.00222\left(\frac{t_s + t}{2}\right) - 0.116 \tag{19}$$

A careful review of Grindley's work has been made by Professor Reeve, of Lawrence Scientific School, who has recalculated the values as originally given. Grindley's results, as recalculated by Reeve, and Greissmann's values, as given by Orrok's formulas, agree quite closely. It is to be noted that Greissmann's work indicated that the specific heat increases with the temperature without regard to what the pressure is; that is, steam of low pressure and high superheat would have the same specific heat as steam of high pressure and low superheat, provided the temperatures were the same. The conclusion was drawn by Lorenz, however (London *Engineer,* July 8, 1904), that specific heat increases with increase in pressure, but decreases with increase of superheat at any given pressure. This conclusion is confirmed by later experiments which will be referred to.

Results of Knoblauch, Linde and Klebe.—In the Stevens Institute *Indicator* for October, 1905, is an article by Prof. J. E. Denton, referred to in the last footnote, outlining the remarkable work done by these investigators. Their work was mainly the finding of the volume of saturated and superheated steam under different pressures and temperatures; but from the equation connecting these elements they were able to deduce equations for the specific heat of superheated steam, and also to explain the reasons

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for the incongruities in the results of some of the previous researches. A table prepared by Professor Denton from their formula is given below:

MEAN SPECIFIC HEAT AT CONSTANT PRESSURE.
(KNOBLAUCH, LINDE, AND KLEBE.)

Boiler Pressure Lb. Sq. In. Absolute.	Boiling Point Degrees C.	Range of Superheating. 10° C. — 50° F.	50° C. — 122° F.	100° C. — 212° F.
99.48	164	0.567	0.551	0.537
189.82	178	.597	.577	.559
190.70	192	.634	.609	.586
266.20	208	.686	.656	.626

Results at Cornell University.—Experiments upon specific heat of superheated steam have been under way for over 10 years at Cornell University. The following table is made up from a chart giving results obtained by Prof. Carl C. Thomas and Mr. C. E. Burgoon at this university, and published by Prof. R. C. Carpenter in a paper, "Steam Plant of the White Motor Car," read before the A. S. M. E. in December, 1906.

SPECIFIC HEAT AT CONSTANT PRESSURE.
(THOMAS AND BURGOON.)

Pressure Lb. Sq. In. Abs.	Degrees F. Superheat. 25	50	100	150	200
50	.515	.513	.512	.510	.509
100	.549	.544	.54	.536	.530
150	.581	.576	.57	.561	.554
200	.614	.61	.598	.589	.577
250	.645	.642	.629	.618	.602

Specific Volume of Superheated Steam.

While the specific volumes of saturated steam for different pressures are to be found in the steam tables, if calculations are to be made requiring the specific volume of superheated steam, the information is not so readily obtained.

Zeuner's Formula.—The formula usually employed is that of Zeuner, based upon the experiments of Hirn. It is as follows:—

$$v = \frac{93.5T_s - 971P^{\frac{1}{4}}}{P} \tag{20}$$

In this P is pressure in pounds per square foot, or $144 \times p$.

Example.—Superheated steam, having a pressure of 100 pounds absolute and a temperature of 400 degrees F., or 860.7 degrees absolute, has a specific volume of

$$v=\frac{93.5\times860.7-971(144\times100)^{\frac{1}{4}}}{144\times100}$$

$$=\frac{80{,}475-10{,}635}{14{,}400}$$

$$=4.8 \text{ cubic feet.}$$

Schmidt's Formula.—Another formula that has been proposed is that of Schmidt, given below, which closely approximates Hirn's results, though not as closely as Zeuner's formula. The difference between the results obtained with the two formulas is slight, however.

$$v=0.59276\frac{441.4+t_s}{p} \qquad (21)$$

Example.—Taking the same data as above, we have,

$$v=0.59276\frac{441.4+400}{100}$$

$$=4.98 \text{ cubic feet.}$$

CHAPTER XIII

CALCULATIONS ON THE FLOW OF STEAM.

The Adiabatic Flow of Steam.

In calculations on the flow of steam it is assumed that the flow is adiabatic and afterwards allowances are made, if necessary, based upon the results of actual tests. Under this condition all the available heat energy of the steam is assumed to be converted into kinetic energy, without gain or loss of heat through conduction, radiation, friction or otherwise, and the energy of the steam will remain the same in amount at all steps in the process, though it may differ in form.

Equation for the Flow of Saturated Steam.—In Fig. 1 is a cylinder having a diaphragm which separates it into two cham-

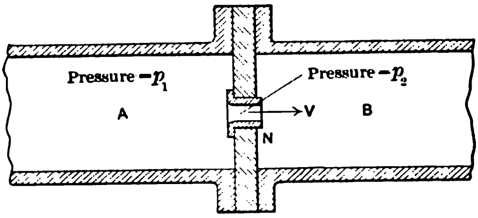


Fig. 1.

bers, A and B. A nozzle N is inserted in the diaphragm and steam in chamber A, at absolute pressure p_1, flows through the nozzle into chamber B. Steam expands within the nozzle to the pressure p_2.

Let V=velocity of steam in feet per second as it leaves the nozzle. Also, let x_1, r_1 and q_1 apply to steam at the pressure p_1, in chamber A, and x_2, r_2 and q_2 to steam at pressure p_2, within the nozzle. (Notation at beginning of Chap. XII.)

Now, since the energy remains constant during the flow, we write expressions for the energy of one pound of steam as it approaches the nozzle, and for one pound of steam as it leaves the nozzle, and place one equal to the other.

The energy of the steam as it approaches or leaves the nozzle is in the form either of heat energy or kinetic energy, the latter due to its velocity. Usually the velocity of approach is so low that the corresponding kinetic energy may be neglected, and we will assume that in chamber A we have to deal with heat energy only. In turbine work the initial velocity is sometimes a considerable factor, however, and in such a case should be taken into account.

Heat energy of one pound of steam in chamber A is $x_1 r_1 + q_1$ heat units, equivalent to $J(x_1 r_1 + q_1)$ foot pounds. (3) Chap. XII.

Heat energy of one pound of steam as it leaves the nozzle is $x_2 r_2 + q_2$ heat units, equivalent to $J(x_2 r_2 + q_2)$ foot pounds.

Kinetic energy of one pound of steam as it leaves the nozzle is, by the principles of mechanics,

$$\frac{V^2}{2g} \text{ foot-pounds.}$$

Placing the energy of discharge equal to the energy of approach,

$$\frac{V^2}{2g} + J(x_2 r_2 + q_2) = J(x_1 r_1 + q_1), \tag{1}$$

which is the general equation for the flow of saturated steam.*

The mechanical energy of the jet in foot-pounds is

$$\frac{V^2}{2g} = J(x_1 r_1 - x_2 r_2 + q_1 - q_2). \tag{2}$$

The velocity of discharge in feet per second is

$$V = \sqrt{2gJ(x_1 r_1 - x_2 r_2 + q_1 - q_2)} \tag{3}$$

$$= 8.02\sqrt{778(x_1 r_1 - x_2 r_2 + q_1 - q_2)} \tag{4}$$

Value of x_2 in Above Equations.—In order to use these equations, we must first calculate x_2, the percentage of dry steam at the end of adiabatic expansion in each case. To obtain x_2 use the

*There should also be included as part of the energy of the steam at the initial pressure, the work required to pump one pound of water from the final pressure p_2 to the initial pressure p_1. This is represented by the expression

$$\frac{144}{62.4}(p_1 - p_2),$$

and is so small in amount that it can be neglected.

characteristic equation (15) Chap. XII., which, when transposed, becomes

$$x_2 = \frac{\theta_1 - \theta_2 + \dfrac{x_1 r_1}{T_1}}{\dfrac{r_2}{T_2}} \qquad (5)$$

If steam tables containing values of θ are not obtainable, it will be necessary to use the approximate equation (10) Chap. XII., for finding the entropy of water in the above equation.

Example I.—Given, dry saturated steam flowing from a pressure of 135 pounds absolute to a pressure of 45 pounds absolute. Calculate the energy of the jet and the velocity of flow, assuming complete expansion in the nozzle.

The following values are either known or taken from the steam table:—

$p_1 = 135$	$p_2 = 45$
$T_1 = 810.73$	$T_2 = 734.99$
$r_1 = 867.3$	$r_2 = 922$
$q_1 = 321.4$	$q_2 = 243.6$
$\theta_1 = .5027$	$\theta_2 = .4020$
$x_1 = 1$	$x_2 = ?$

First, calculate the percentage of dry steam at the end of the expansion, from equation (5).

$$x^2 = \frac{.5027 - .4020 + \dfrac{1 \times 867.3}{810.73}}{\dfrac{922}{734.99}}$$

$$= .933 = 93.3\%.$$

Now substituting for x_2 in equation (2), we have, for energy of the jet,

$$\frac{V^2}{2g} = 778(1 \times 867.3 - .933 \times 922 + 321.4 - 243.6)$$

$$= 66{,}032 \text{ ft. lb.}$$

and velocity of discharge is

$$V = \sqrt{64.4 \times 66{,}032}$$

$$= 2{,}062 \text{ ft. per sec.}$$

When Expansion is not Complete in the Nozzle.—In working out the above example it was assumed that the expansion of the steam was carried to the terminal pressure of 45 pounds within the nozzle itself and there was no waste energy due to drop of pressure as the steam left the nozzle. To accomplish this with the pressures given requires a diverging nozzle, as already explained in Chapter I. under "Steam Nozzles," and in Chapter XI. Suppose, however, that instead of a diverging nozzle a straight nozzle with converging inlet were used. We have learned that in such a nozzle expansion may be carried to a certain point—usually about 60 per cent of the higher absolute pressure—*and no further*, and in this case steam would expand within the nozzle to about 80 pounds, which should be used for the lower pressure p_2 in the calculations. *In any case where expansion is not complete within the nozzle, care should be taken to assume for p_2 the pressure to which steam expands within the nozzle itself instead of the lower outside pressure.*

Example II.—Assuming a straight nozzle, we have for values corresponding to $p_2=80$ pounds, $r_2=895.6$, $q_2=281.4$, $\theta_2=0.452$, $T_2=772.5$.

Values corresponding to p_1 are given under Example I.

From (5), we find $x_2=0.966$.

From (3),

$$V=\sqrt{64.4\times778(867.3-.966\times895.6+321.4-281.4)}$$
$$=1{,}451.4 \text{ ft. per sec.}$$

Chart Giving Values of x.—To assist the reader in determining the percentage of dry steam at the end of adiabatic expansion, the chart in Fig. 2 has been prepared, which enables the value of x_2 to be read directly, without calculation. This quantity may also be easily determined by the aid of the temperature-entropy diagram, as explained in connection with the subject in Chapter XII.

Simplified Formula for the Flow of Steam.—In the steam table in the appendix values of the entropy, ϕ, of steam are given, in which

$$\phi=\frac{r}{T}+\theta$$

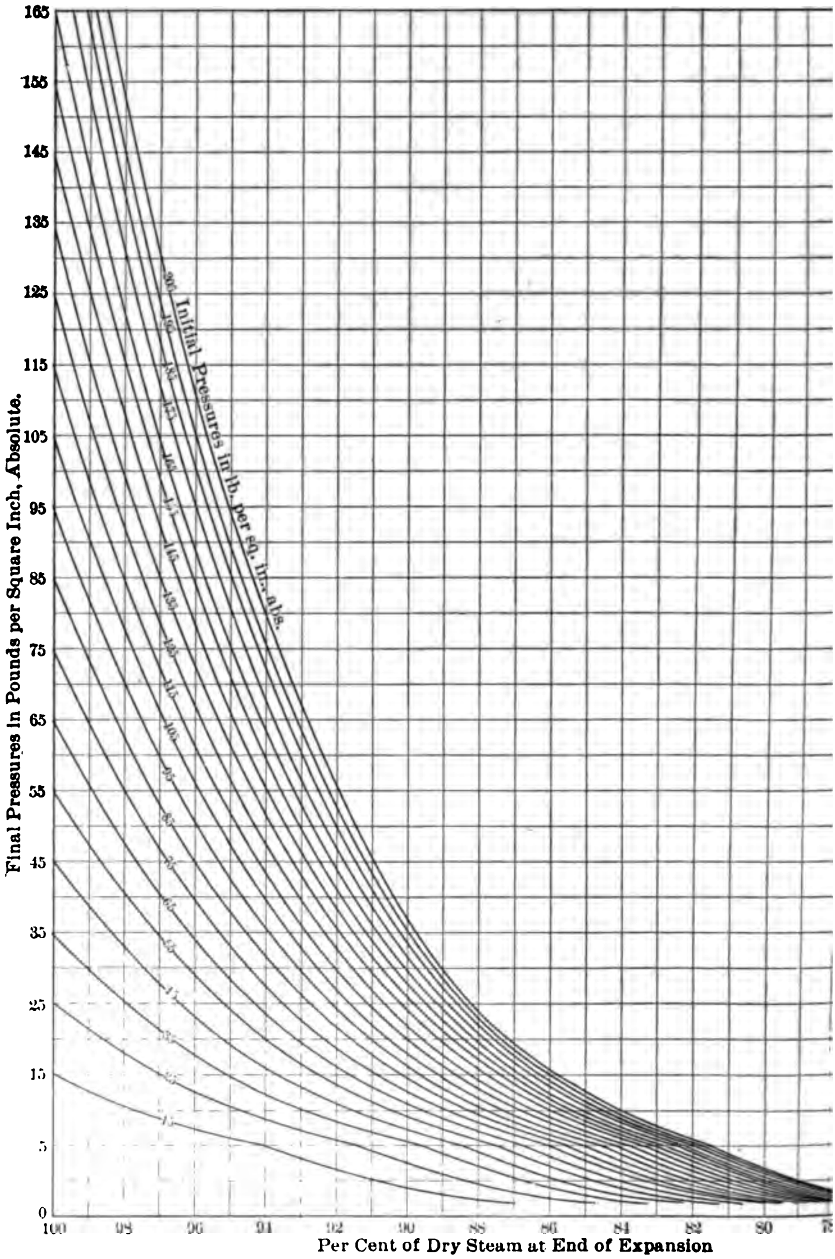


Fig. 2. Chart Showing Per Cent of Dry Steam Present at the End of Adiabatic Expan

Hence, if, as is usually assumed in calculations, the steam is initially dry, equation (5) reduces to

$$x_2 = \frac{T_2(\phi_1 - \theta_2)}{r_2} \tag{6}$$

Also, with steam initially dry, we may write λ_1, the total heat of dry steam, in place of $(x_1 r_1 + q_1)$ in (2), which gives,

$$\frac{V^2}{2g} = J(\lambda_1 - x_2 r_2 - q_2) \tag{7}$$

Now substituting in this the value for x_2 in (6), we get,

$$\frac{V^2}{2g} = J[\lambda_1 - T_2(\phi_1 - \theta_2) - q_2] \tag{8}$$

This gives the flow directly without computing x_2.

Example III.—Assume expansion complete within the nozzle.

$p_1 = 185$	$p_2 = 15$
$\lambda_1 = 1{,}196.4$	$T_2 = 673.73$
$\phi_1 = 1.5498$	$\theta = .3143$
	$q_2 = 181.8$

$$\frac{V^2}{2g} = 778\ [1{,}196.4 - 673.73(1.5498 - .3143) - 181.8]$$

$$= 141{,}757.$$

$$V = \sqrt{64.4 \times 141{,}757}$$

$$= 3{,}020 \text{ ft. per sec.}$$

Weight of Saturated Steam Flowing.—The most satisfactory, as well as the simplest equations for the weight of steam discharged from a nozzle are Napier's empirical formulas given at the beginning of Chapter XI. In the design of nozzles, however, it may sometimes be more convenient to use the equations (9 and 10) below, which, instead of being based upon experiment, like Napier's, depend upon the following principle:—

Weight in pounds discharged per second × volume in cubic feet of one pound (specific volume) = area of orifice in square feet × velocity of steam in feet per second.

Let W=weight discharged per second in pounds.
v=specific volume of the fluid in the orifice.
A=area of orifice in square feet.
V=velocity of discharge in feet per second.

Then,
$$W \times v = A \times V$$
$$W = \frac{AV}{v} \qquad (9)$$

This formula is perfectly general and applies either to saturated or superheated steam, or to any gas or liquid.

If it is to be applied to saturated steam, then, by (5), Chap. XII., we have, $v = xs$, where x is the quality and s is the specific volume of the steam in the orifice. Also,

$$A = \frac{a}{144}$$

where a is area in square inches.

Hence,
$$W = \frac{aV}{144xs} \qquad (10)$$

Example IV.—What is the weight of steam discharged in Example II., assuming the area of nozzle to be 0.5 square inch?

Here we have, initial pressure 135 pounds, and pressure in nozzle $\frac{6}{10}$ of this, or approximately 80 pounds. The quality x at this lower pressure was found to be 0.966 and the velocity V 1,451 feet per second. The specific volume s corresponding to 80 pounds is 5.43.

Hence, from (10),

$$W = \frac{.5 \times 1{,}451}{144 \times .966 \times 5.43}$$
$$= 0.961 \text{ pounds per second.}$$

Napier's rule gives 0.964 as the result.

Calculations Upon Superheated Steam.

Equations for the Flow of Superheated Steam.—There are two cases according as the steam is superheated or saturated when expansion is completed, its condition at that point depending upon the degree of superheat at the start and the degree of expansion that takes place. The calculation of the velocity or energy of

flow of superheated steam is a long and tedious process, and until the results of more tests are available we are not sure that the calculated results agree even approximately with experimental results. Nearly all tests have so far been upon saturated steam.

The formulas for the flow of superheated steam are derived by the same method as the one for saturated steam.

The heat energy of one pound of superheated steam is

$$\lambda + c\,(t_s - t) \qquad (6) \text{ Chap. XII.}$$

Placing the energy of discharge equal to the energy of approach, and using letters with subscripts 1 and 2 to represent initial and final conditions, respectively, we have:

Case I., when steam is superheated at the end,

$$\frac{V^2}{2g} + J\,[\lambda_2 + c\,(t_{s2} - t_2)] = J\,[\lambda_1 + c\,(t_{s1} - t_1)]$$

or,

$$\frac{V^2}{2g} = J\,[\lambda_1 - \lambda_2 + c\,(t_{s1} - t_1) - c(t_{s2} - t_2)] \qquad (11)$$

Case II., when steam is saturated at the end,

$$\frac{V^2}{2g} + J\,(x_2 r_2 + q_2) = J\,[\lambda_1 + c\,(t_{s1} - t_1)]$$

or,

$$\frac{V^2}{2g} = J\,[\lambda_1 + c\,(t_{s1} - t_1) - x_2 r_2 - q_2] \qquad (12)$$

To find the Pressure at which Superheated Steam Loses its Superheat During Adiabatic Expansion.—Before one can proceed to calculate the velocity of flow of superheated steam it must be determined whether the example comes under Case I. or Case II.; that is, whether the steam is superheated or saturated at the end of the flow. This is done by making $x_2 = 1$ in characteristic equation (17), Chap. XII., for superheated steam. The second member of this equation will then be

$$\frac{r_2}{T_2} + \theta_2 = \phi_2,$$

which is the entropy of dry, saturated steam, and the equation will express the relation that the entropy of superheated steam at a

certain pressure and temperature equals the entropy of dry, saturated steam at a certain pressure, which latter pressure is to be determined. After making $x_2=1$, find the value of the left-hand member of the equation, all the quantities of which have known values. The result will be the entropy of saturated steam for a certain corresponding pressure, to be obtained from the steam tables. This pressure will be that at which the steam gives up its superheat for the example in question and will indicate whether the steam is superheated or saturated at the end of the flow. If the pressure is greater than the final pressure against which the steam is flowing, it shows that the steam becomes saturated before the final pressure is reached, and hence will be saturated at the end. If the pressure determined is less than the final pressure, the steam will be superheated at the end.

Chart Showing Pressures at which Superheated Steam Gives Up Its Superheat.—To facilitate calculations the chart in Fig. 3 has been calculated by which the pressure at which superheated steam gives up its superheat in adiabatic expansion can be read off directly and the condition of the steam at the end of the expansion determined. Each curved line is for a different initial pressure. The pressures at the left are those at which steam superheated a given amount gives up its superheat and becomes saturated. The vertical lines correspond to different degrees of superheat.

Calculation of the Velocity of Flow of Superheated Steam.—Having found whether the given problem must be solved by equation (11) under Case I. or equation (12) under Case II., proceed as follows:—

Case I.—The steam in this instance is superheated at the end of the flow. All the quantities of equation (11) will be known or can be obtained from the steam table except t_{s2}, the final temperature of the superheated steam. This must be calculated by the aid of characteristic equation (16), Chap. XII., for superheated steam.

Case II.—Here the steam is saturated at the end of its flow and equation (12) will be used. All the quantities of this equation are known except x_2, which is to be obtained from characteristic equation (17), Chap. XII., for superheated steam.

Example V.—Steam, superheated 100 degrees F., flows adiabat-

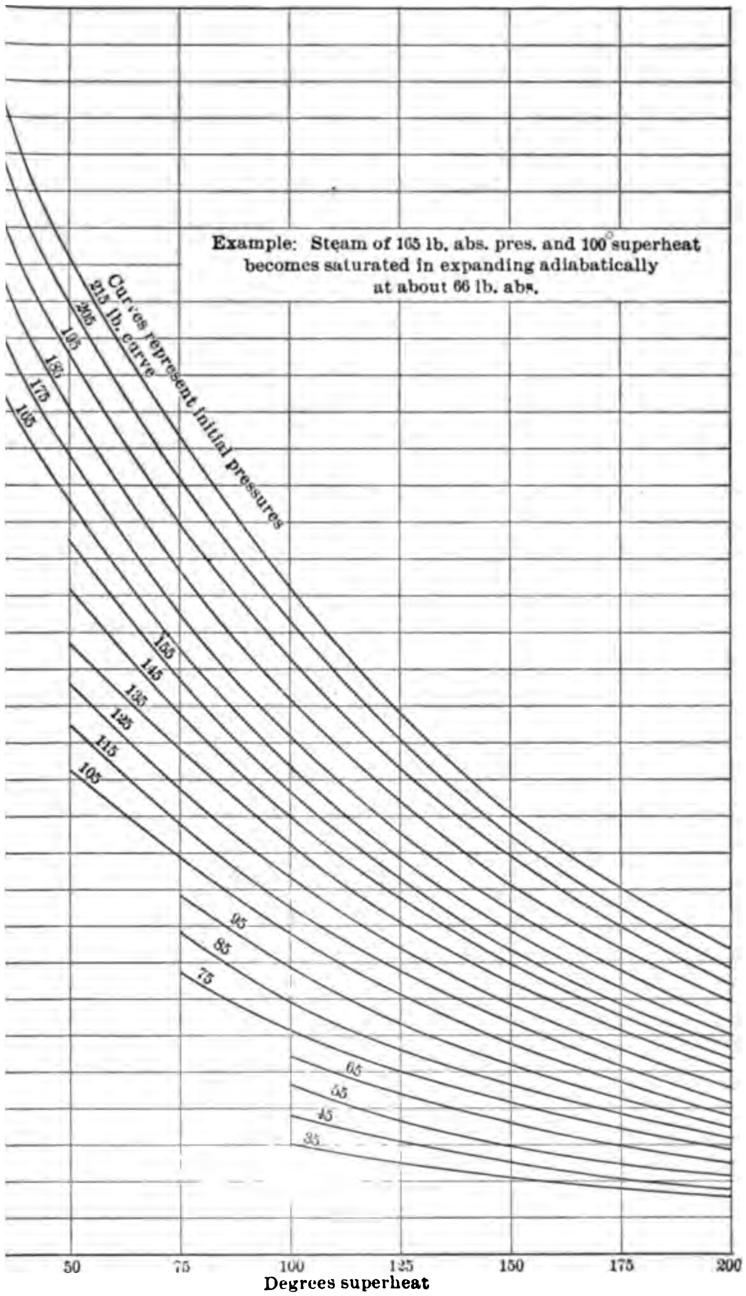


Chart Showing Pressures at which Superheated Steam Gives up its Superheat in Adiabatic Expansion. Specific Heat assumed=0.6.

ically from a pressure of 135 pounds absolute to a pressure of 45 pounds absolute. What is its velocity of discharge? Assume specific heat, c, to be 0.6.

From the diagram, Fig. 3, we find that steam superheated 100 degrees and flowing from a pressure of 135 pounds to a pressure of 45 pounds will give up its superheat at 56 degrees and hence be saturated at the end of the flow, making the example come under Case II. We find x_2 from equation (17), Chap. XII., as follows:—

$$x_2 = \frac{T_2}{r_2}\left(\frac{r_1}{T_1} + \theta_1 + 2.3\,c\log\frac{T_s}{T_1} - \theta_2\right)$$

$$= \frac{734.99}{922}\left[\frac{867.3}{810.73} + .5027 + \left(2.3 \times 0.6 \times \log\frac{910.73}{810.73}\right) - .402\right]$$

$$= 0.988.$$

From equation (12) the velocity becomes,

$$V = \sqrt{2g \times 778\left[\lambda_1 + c\,(T_{s1} - T_1) - x_2 r_2 - q_2\right]}$$

$$= \sqrt{50{,}103\,(1188.7 + 60 - 911.46 - 243.6)}$$

$$= 2166 \text{ ft. per sec.}$$

This velocity is only slightly greater than calculated for saturated steam flowing between the same pressures in Example I. The slight increase is due to the additional heat in the superheat, but to partially offset this there is more heat carried away at the end, in the form of latent heat, since in this example the steam is more nearly dry at the end than in the previous example.

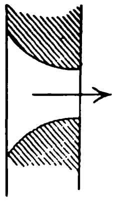
STEAM NOZZLE DESIGN.

The fundamental (and self-evident) principle upon which the design of steam nozzles is based is that the different cross-sectional areas of the nozzle must be sufficient to allow a given weight of the fluid to pass in a given space of time. In other words, the same weight of fluid must pass different sections of the nozzle in the same time.

This relation is expressed by formula (9) for the weight of flow,

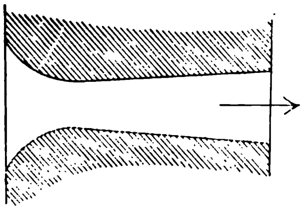
$$W = \frac{AV}{v}$$

The Reason for Converging and Diverging Nozzles.—Let us examine this formula, and assume, first, that a *liquid* is flowing through the nozzle. The specific volume v of a liquid is constant and hence, as the velocity V increases, owing to drop in pressure, the area A must decrease; or in other words, the nozzle will converge, as in Fig. 4.



CONVERGING

Fig. 4.



CONVERGING AND DIVERGING

Fig. 5.

Again, suppose *steam* to flow through the nozzle. In this case the specific volume, as well as the velocity, will increase with the drop in pressure. It is a characteristic of steam that at first the velocity increases more rapidly than the specific volume and later the specific volume increases more rapidly than the velocity; and to conform to these conditions the nozzle should first converge and then diverge, as in Fig. 5.

Critical Pressure.—The point where the ratio

$$\frac{V}{v}$$

changes from an increasing to a decreasing quantity is called the critical point, and theoretically is at a pressure of 0.58 of the initial pressure.* The pressure at the throat is therefore 58 per cent of the higher pressure, theoretically, although tests in Chapter XI. show that it may vary materially from this in certain nozzles.

*Professor Rateau of Paris, in *Flow of Steam Through Nozzles*, states that he has calculated the points at which the ratio

$$\frac{V}{v}$$

becomes a maximum for a number of different initial pressures and finds they vary slightly with the pressure around the value $0.58p$. It is a very peculiar fact that this relation exists so closely for different pressures. Mr. Joseph C. Riley, Massachusetts Institute of Technology, Boston, has shown that the value is affected somewhat by moisture in the steam. For 100 pounds initial pressure and 20 per cent priming the pressure at which the ratio changes is 0.54—not a wide variation.

Converging Nozzle for Steam.—In case expansion is not carried far enough to pass the critical point, where

$$\frac{V}{v}$$

begins to decrease, the diverging section of the nozzle is not required, a condition that exists when the discharge pressure is 58 per cent or more than 58 per cent of the initial pressure, as was fully explained in Chapter I. In this case the nozzle of Fig. 4 answers all requirements.

Method of Procedure in Nozzle Design.—Calculations for superheated steam are so complicated that the method and calculations for saturated steam only will be presented here. In designing for superheated steam the same objects are to be attained, but we have to use the superheated steam formulas for specific volume, velocity, etc., in so far as such formulas have been developed and are available.

In steam turbine design we know in advance how many foot-pounds of energy per second we wish to have delivered to the wheel by the steam, and what drop in pressure there is to be in flowing through the nozzle. Equation (2) gives the energy developed per pound of steam for a given drop of pressure, and the total energy required divided by the energy per pound will give the number of pounds of steam per second, or the fraction of a pound, as the case may be, that the nozzle must be able to deliver.

From this, the area of the nozzle can be determined by the aid of equation (9). This equation may be written

$$A=\frac{Wv}{V} \tag{13}$$

or

$$a=\frac{144Wv}{V} \tag{14}$$

If the nozzle is of circular cross-section its diameter in inches is given by

$$d=\sqrt{\frac{576}{\pi}\times\frac{Wv}{V}} \tag{15}$$

Or, if the designer prefers, he may use in place of these equations those of Napier for the flow of steam, in Chapter XI., with as good or better results. The calculation of the area of a straight or converging nozzle is so simple that no example will be required.

Design of a Diverging Nozzle.—In a diverging nozzle we have in addition to the area of the throat to find the larger outlet area.

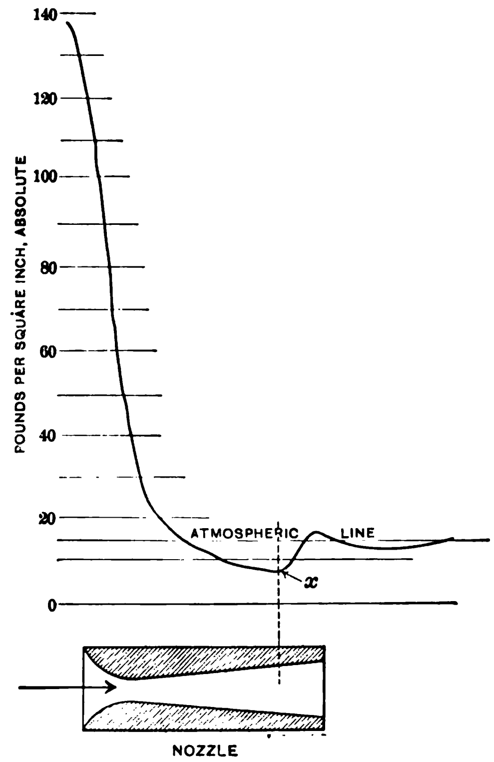


Fig. 6. Nozzle with Divergence Carried too Far.

It has previously been pointed out, in connection with the tests of Chapter XI., and elsewhere, that if the divergence of the nozzle is carried too far, the pressure of the steam will rise above the terminal outside pressure before leaving the nozzle and there will be an attendant loss of energy. This is also shown in Fig 6, which is a record of an experiment by Francis Hodgkinson, and reported in a pamphlet issued by the Westinghouse Machine Company.

The nozzle discharges against atmospheric pressure, but the divergence of the nozzle is so great that the steam reaches a pressure corresponding to that of the atmosphere shortly after the throat is passed; and beyond this point the steam expands to considerably below atmospheric pressure, reaching the lowest point at x. Then, as it discharges to the atmosphere, the pressure rises, and, of course, the velocity decreases. This shows the importance of securing a correct relation between the throat and outlet areas of a diverging nozzle. The method to be followed can best be shown by an example.

Example Illustrating the Design of a Diverging Nozzle.—Required, the throat and outlet areas for a nozzle to deliver one pound of steam per second, flowing from a pressure of 135 pounds absolute to a pressure of 45 pounds absolute.

First, consider the throat area on line *aa*, Fig. 7. The steam, in flowing through the converging inlet of the nozzle, will expand to about $\frac{6}{10}$ of the absolute initial pressure, or to a pressure of 80

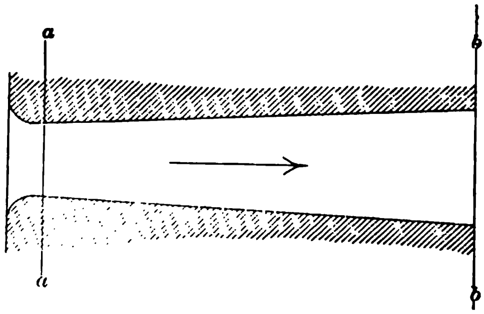


Fig. 7.

pounds, approximately. In Example II., where the steam expanded from 135 to 80 pounds, as in the present case, the velocity V was found to be 1,451.4 feet per second. In examples where steam is known to expand to $\frac{6}{10}$ of the higher pressure, however, it is not necessary to calculate the velocity, since for all ordinary initial pressures the velocity will be practically constant and range close to 1,450 feet per second. (See Table I., Chapter XI.) From Example II., also, for pressure 80 pounds, $x=0.966$, while s, from steam table$=5.43$.

Hence, for section *aa* of the nozzle,

$p=80$, $V=1{,}451$, $W=1$, and $v=xs=0.966\times 5.43=5.245$.

From (14),

$$a=\frac{144\times 1\times 5.245}{1{,}451}$$

$$=0.52 \text{ square inch.}$$

Second, consider the outlet area on line *bb*. Here the absolute pressure is 45 pounds and in Example I., where steam expanded from 135 pounds to 45 pounds, as in this case, we found $V=2{,}062$ feet per second and $x=0.933$. From the steam tables, s for 80 pounds$=9.29$.

Hence, for section *bb* of the nozzle,

$p=45$, $V=2{,}062$, $W=1$, and $v=xs=0.933\times 9.29=8.667$.

$$a=\frac{144\times 1\times 8.667}{2{,}062}$$

$$=0.605 \text{ square inch.}$$

It will be evident from the above that the two areas are directly as the specific volumes and inversely as the velocities.

Practical Considerations.

As stated at the outset, the calculations in this chapter are based upon the assumption of adiabatic flow. It is a peculiar fact, however, that while such calculations approximate closely to the results of tests, it is quite certain that the flow through nozzles is not adiabatic, or at best it is only approximately so. This is indicated by the tests of Professor Lucke in Chapter XI., and is further shown by the fact, which any one can verify, that the steam discharging from a nozzle does not contain the amount of moisture called for by adiabatic expansion. It is the experience of experimenters that the discharge is either blue, indicating dry steam, or dry and white, indicating not over two per cent of moisture.

Frictional Losses.—Nozzles of different shapes and proportions have different coefficients of flow and the best the designer can do in allowing for frictional losses is to select his own co-

efficients from tests upon nozzles as nearly as possible like those he contemplates using. It is apparent from the tests of Chapter XI. that a converging nozzle can be designed to give a velocity of discharge within two per cent of the theoretical velocity. The diverging nozzles show a wider variation, and in some cases a very wide variation, depending upon their proportions. In calculating the weight of steam discharged Napier's rules can be used, as already explained, but a proper coefficient must be selected in each case from tests upon similar nozzles.

Diverging Nozzles.—Rosenhain concludes from his experiments that the taper of diverging nozzles should not be far from 1 in 12 and that the inner edge of the nozzle should be only slightly rounded. De Laval nozzles are made with tapers ranging from about 1 in 10 to 1 in 20, with inlet only slightly rounded. The experiments of Mr. Strickland L. Kneass, engineer of the injector department of William Sellers & Co., Philadelphia, indicate that better results are obtained from nozzles with well-rounded inlets and a taper of about 1 in 6; and that if properly proportioned it is possible to secure a velocity of flow within two per cent of theoretical. One of several series of experiments, records of which have been furnished the author by Mr. Kneass, was upon five different nozzles having a divergent taper of 1 in 6, but with a ratio of discharge to throat areas carefully calculated for different initial pressures. The nozzles discharged at atmospheric pressure against a parabolic target, which deflected the steam through an angle of 90 degrees. The target was connected with a delicate weighing device by which the impact of the steam could be accurately determined.

Of the several nozzles, one designed for 30 pounds initial pressure, gauge, and having a throat diameter of 4.14 mm and discharge diameter of 4.32 mm, gave the best average results throughout the whole range of pressures, as per table below:—

Initial Pressure, Gauge.	Impact Pressures in lb. per sq. mm of Nozzle.	Weight Discharged, lb. per sq. mm of Nozzle.
120	0.2401	0.002870
90	0.1783	0.002249
60	0.1174	0.001648
30	0.0591	0.000965
15	0.0299	0.000688

From these results the author has calculated the actual velocities of discharge, using the formula employed in Rosenhain's tests, and compared them with the theoretical velocities, taken from Chart No. 3 in the appendix, as follows:—

Gauge Pressure.	Actual Velocity.	Theoretical Velocity.	Difference.	Loss, Per Cent.
120	2,690	2,820	130	.046
90	2,550	2,650	100	.038
60	2,290	2,400	110	.045
30	1,970	2,000	30	.015
15	1,400	1,500	100	.066

These results show that at 30 pounds, for which the nozzle was designed, the loss is only 1.5 per cent, and by comparing results for the whole range of pressures it will be evident that a nozzle is more efficient when designed for too low a pressure than for too high a pressure.

From the velocities given above the actual and theoretical energy of the steam jet may be calculated and compared. The energy loss in the above cases will be found to range from 3 to 13 per cent.

CHAPTER XIV

TURBINE VANES.

The Vanes of Impulse Turbines.

In preceding chapters we have studied the principles of the flow of steam and the conversion of the heat energy of steam into the mechanical kinetic energy of the escaping jet. The subject now to be considered is the transference of this mechanical energy of the jet to the vanes of the turbine wheel.

At the beginning of Chapter I. the meaning of absolute and relative motion was explained, as related to the action of a fluid upon a moving vane, and the treatment there given will furnish sufficient introduction for what is to follow.

Diagram for a Moving Vane.—In Fig. 1 herewith a turbine vane moves in the direction of the horizontal arrow and is acted upon by a jet of steam flowing in the direction of the inclined

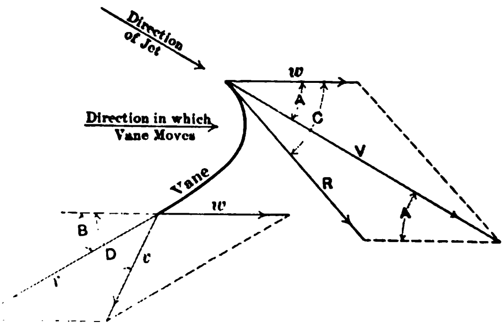


Fig. 1.

arrow. The parallelograms of motion show the velocity and direction of motion of the steam in entering upon and leaving the vane; and the velocity and direction of motion of the vane itself. The lengths of the several lines represent velocities in feet per second, drawn to any convenient scale. In the diagram,

V is the initial and v the final absolute velocity and direction of the steam;

R is the initial and r the final velocity of the steam relative to the vane.

w is the velocity and direction of motion of the vane.

If there were no loss through friction or eddying, the relative velocities R and r would be equal, and this will be assumed the case in what follows, unless otherwise stated.

A is the angle made by V with the direction of motion of the vane; C, the angle made by R; and D and B, the corresponding angles made by the steam when leaving the vane. In what follows A and D will be designated as the "initial" and "final" angles, and C and B the "entrance" and "exit" angles.

For tangential action upon the vane, allowing the steam to enter upon it without impact and to leave it without commotion, the vane should be tangent to R at the entrance and tangent to r at the exit. The shape of the vane between the entrance and exit is not very important, so long as the curve is gradual and smooth.

In hydraulic turbines the water usually flows either outward or inward, in a direction generally radial; and as the vanes are large it is necessary to take into account the difference in velocity of their inner and outer circumferences when proportioning the angles, etc. In steam turbine work this is not necessary, since the steam usually flows in an axial direction and it is sufficiently accurate to assume that the vane moves forward in a straight line at a speed equal to that of the mean circumferential speed of the vanes.

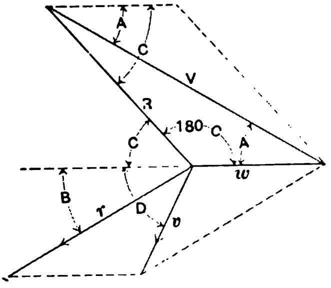


Fig. 2.

Calculating the Parts of the Diagram.—Fig. 2 shows a convenient arrangement of the velocity diagram for finding the values of the different elements, either by graphical construction or by

calculation. The several lines are lettered to correspond with Fig. 1 and the several parts can be calculated by the simple formulas of trigonometry. The most important formula used is the one stating that "In any triangle the square of any side is equal to the sum of the squares of the other two sides, minus twice their product into the cosine of their included angle."

For example, if there are given values of V, w, and angles A and B; and it is required to find R ($=r$), v and angles C and D, then

$$R^2=w^2+V^2-2\,w\,V\cos A \qquad (1)$$

$$v^2=r^2+w^2-2\,r\,w\cos B \qquad (2)$$

Also, it can be shown that

$$\operatorname{Cos} C=\frac{V\cos A-w}{R} \qquad (3)$$

$$\operatorname{Cos} D=\frac{r\cos B-w}{v}$$ (when line v is inclined to the right as in Fig. 2). (4)

$$\operatorname{Cos} D=\frac{w-r\cos B}{v}$$ (when line v is inclined to the left). (5)

Turbine Efficiency.—The energy of one pound of steam impinging against the turbine vanes is

$$\frac{V^2}{2g};$$

and of one pound of steam as it leaves the vanes is

$$\frac{v^2}{2g}$$

The energy absorbed by the vane is therefore

$$\frac{V^2-v^2}{2g}$$

By the principle of machines,

$$\text{Efficiency}=\frac{\text{energy absorbed by vanes}}{\text{total energy delivered to vanes}}$$

$$=\frac{V^2-v^2}{V^2},$$ the $2g$ canceling in each case so that velocities only need be considered.

But by trigonometry,

$$V^2 = w^2 + R^2 - 2\ w\ R \cos\ (180 - C),$$

which reduces to

$$V^2 = w^2 + R^2 + 2\ w\ R \cos\ C.$$

Also,

$$v^2 = w^2 + r^2 - 2\ w\ r \cos\ B,$$

whence,

$$\frac{V^2 - v^2}{V^2} = \frac{w^2 + R^2 + 2\ w\ R \cos C - (w^2 + r^2 - 2\ w\ r \cos B)}{V^2} \tag{7}$$

$$= \frac{R^2 - r^2 + 2\ w\ (R \cos C + r \cos B)}{V^2} \tag{8}$$

Assuming that $R = r$, as previously explained,

$$\text{Efficiency} = \frac{V^2 - v^2}{V^2} = \frac{2\ w\ R\ (\cos C + \cos B)}{V} \tag{9}$$

Conditions of High Efficiency.—It is obvious that to attain high efficiency the final absolute velocity v of the steam must be small, as otherwise energy would be wasted without doing work on the vanes. In Fig. 3 v has a small value and it will be evident that this is partly due to the fact that angle B is small and partly that w and r are equal. A few minutes spent in drawing diagrams will demonstrate that if angle B is large, or if w and r differ considerably in length, the value v will be materially increased.

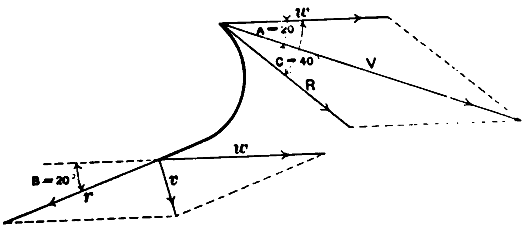


Fig. 3.

In laying out a diagram economical conditions may be attained by selecting angles A and B as small as practicable, say from 15 to 20 degrees, and angle $C = 2A$, thus making w and R equal.

The value of w, and hence of R, will then be calculated by

$$w=\frac{V}{2\cos A} \tag{10}$$

The final absolute velocity is calculated by

$$v=w\sqrt{2\,(1-\cos B)} \tag{11}$$

While these proportions do not give quite the maximum efficiency, they produce a simple construction and are satisfactory if it is possible to use them. Steam velocities are so high, however, that the wheel velocity w must usually be selected from considerations of safety and utility rather than of theory.

The question of efficiency can be made clear, without difficult calculations, by reference to the graphical constructions of Figs. 3, 4, and 5.

Efficiency as Shown by Diagrams.—

Example 1.—Let A and B =20 degrees and V=3,000 feet per second.

In accordance with the last article, let $C=2A=40$ degrees;

$w=\frac{V}{2\cos A}=\frac{3{,}000}{2\times.94}=1{,}600$ feet per second. This is also the value of R.

In Fig. 3 draw w to any suitable scale, to represent 1,600, and V at an angle of 20 degrees with w to represent 3,000. Then draw $R=w$ at an angle of 40 degrees with the latter. Complete the parallelogram.

Now construct the parallelogram for the discharge in the same manner, making $r=w$ and angle B=20 degrees.

v is the absolute velocity of discharge and the vane curve is drawn with the entrance and exit surfaces tangent respectively to R and r.

It will be evident that a wheel with vanes laid out as in Fig. 3 will have a high efficiency.

Example 2.—Fig. 4 has been constructed to show the influence of the initial angle A upon the value of v. As in the previous diagram $C=2A$, $w=R$ and B=20 degrees. But angle A has been made 45 instead of 20 degrees and C 90 instead of 40 degrees.

The result is that w and R and hence r are much greater than before and v in consequence is greater and the wheel will be less efficient. It will be noted, however, that while the size of angle A in Fig. 4 is more than double its size in Fig. 3, the value of v is

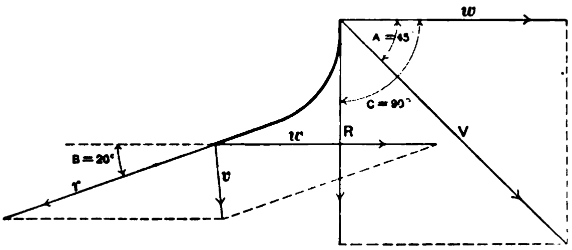


Fig. 4.

increased only a few per cent. There can thus be a considerable latitude in the selection of the initial angle. The chief disadvantage of a large angle in steam turbine work is that it necessitates a high value for the speed w of the wheel circumference; and if we attempt to reduce the wheel velocity and still maintain a large initial angle, the result is not good, as the next example will show.

Example 3.—In Fig. 5 the angle A was made 45 degrees as in the last example, and the angle B 20 degrees, as in both Examples 1 and 2. Instead of selecting $C=2A$, however, the speed w of the wheel was kept the same as in Example 1, or less than in

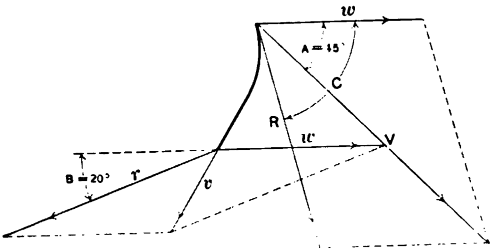


Fig. 5.

Example 2, and the angle C was then determined by drawing the parallelogram. The final result is a value for v greater than in either of the previous cases, indicating that the construction is not so good as where $C=2A$ and $w=R$, as in Figs. 3 and 4.

While the foregoing examples are not of the nature of demonstrations, they indicate why it is desirable to have angle B as small as possible and A reasonably small; while the sides of both parallelograms should be equal, making $C=2A$.

Vanes with Entrance and Exit Angles Equal.—An important case for impulse steam turbines is that of symmetrical vanes having entrance and exit angles equal. With vanes so proportioned there is no thrust to be taken care of, due to the reaction of the steam leaving the vanes, since the reaction is balanced by the impulse of the jet striking the vanes.

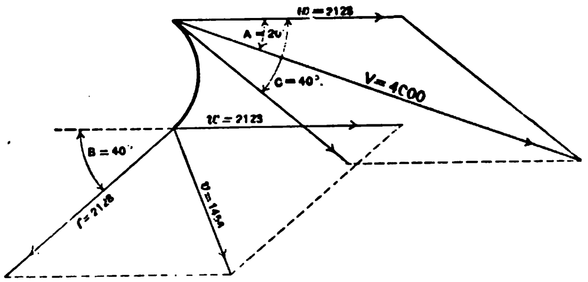


Fig. 6.

In Fig. 6 is the diagram for this construction. Let us assume $V=4{,}000$ feet per second, $A=20$ degrees, $C=2A=40$ degrees (the condition of high efficiency), and B, which is equal to C to make the vane symmetrical, is also 40 degrees.

By formula (10),

$$w=\frac{4{,}000}{2\times .94}=2{,}128 \text{ feet per second.}$$

This is also the value of R and r.

By formula (11),

$$v=2{,}128\sqrt{2\times .234}=1{,}456 \text{ feet per second.}$$

The efficiency may be calculated by either the first or second parts of formula (9) as most convenient. Taking the first part, we have,

$$\text{Efficiency}=\frac{(4{,}000)^2-(1{,}456)^2}{(4{,}000)^2}=86.7 \text{ per cent.}$$

Obviously no such wheel velocity as the above would be possi-

ble and no velocity of steam as great as 4,000 feet per second would usually be attained, although it is sometimes reached in the De Laval turbine. Impulse turbines are ordinarily of the multicellular type and the drop in pressure between any two compartments would not be more than enough to produce the maximum velocity of flow from a converging nozzle, or about 1,450 feet a second. If there were one wheel in each compartment the velocity of its vanes, calculated on the same basis as the last example, would be 771 feet a second, which is within feasible limits.

Formulas for Highest Efficiency.—It has been explained that while the conditions just outlined produce a wheel of high efficiency, they do not produce the highest efficiency possible. In notes issued to his students, Prof. Edward F. Miller, Massachusetts Institute Technology, has worked out formulas for the theoretical highest efficiency for symmetrical vanes. The demonstrations are somewhat difficult and only the results will be given. It is assumed that A and V are known. Then,

$$\text{Tan } C = 2 \tan A \qquad (12)$$

$$w = \frac{V \cos A}{2} \qquad (13)$$

$$v = V \sin A \qquad (14)$$

$$\text{Efficiency} = \cos A \qquad (15)$$

The calculated results for the example of the last article are $C=36$ degrees, plus; $w=1,879$ feet per second; $v=1,368$ feet per second; efficiency$=88.3$ per cent. These may be compared with the results found by the other method.

Efficiency of Pelton Type Wheel.—In Fig. 7 is a Pelton bucket moving in the direction K with a velocity w. The jet strikes the bucket in the direction in which it is moving as indicated by arrow A.

If the bucket were so shaped that there were a complete reversal of the stream, allowing it to escape in the opposite direction from which it entered the bucket, as indicated by the dotted line, the efficiency would, neglecting friction, depend only on the

velocity of the wheel. If the wheel traveled with half the velocity of the jet the efficiency would be 100 per cent, and as the velocity of the wheel decreased the efficiency would grow less.

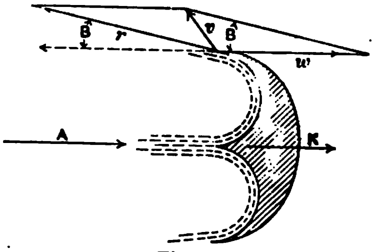


Fig. 7.

This is an impossible condition, of course, and the buckets must be designed to allow the fluid to depart at a slight angle from the path of the bucket, as indicated by r in Fig. 7.

Let w, Fig. 7, be the wheel velocity. If V is the velocity of the jet, then r, the relative velocity of the fluid on the bucket, is $V-w$. The final absolute velocity v is calculated by formula (2),

$$v^2=r^2+w^2-2\ r\ w \cos B,$$

where B is the angle of discharge.

Having found this the efficiency is easily calculated by the first part of the formula (9); or the second part of the formula may be used remembering that angle C is zero and that $\cos 0=1$.

Example.—Let $V=1{,}000$ feet per second; $w=400$ feet; and $B=15$ degrees. Then $r=V-w=600$ feet per second and

$$v^2=360{,}000+160{,}000-2\times600\times400\times.966$$
$$=56{,}320.$$

$$\text{Efficiency}=\frac{1{,}000{,}000-56{,}320}{1{,}000{,}000}=94.4 \text{ per cent.}$$

This, like other calculations for efficiency in this chapter, is purely theoretical and is higher than can be realized in actual conditions.

Diagrams for Compound Impulse Turbines.—In Fig. 8 the initial velocity V_1 of the steam is assumed to be four times the wheel velocity w, and angle $A_1=20$ degrees. The steam, having acquired its velocity, flows of its own momentum through two sets of mov-

ing vanes and one set of guide vanes between them, and finally issues at an absolute velocity v_2. To avoid end thrust the moving vanes must be symmetrical and hence $C_1 = B_1$ and $C_2 = B_2$. If there

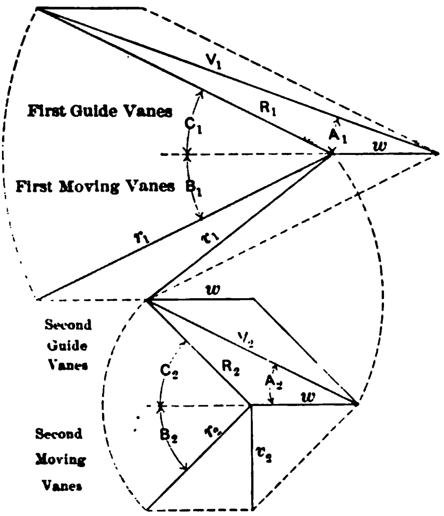


Fig. 8.

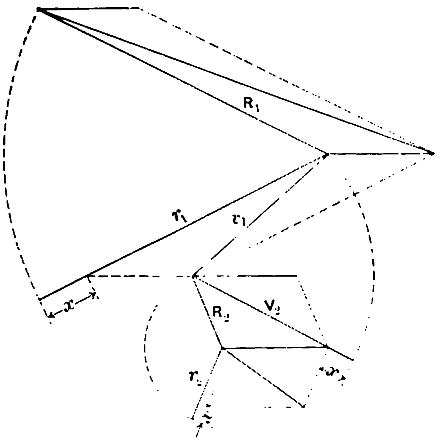


Fig. 9.

were no loss through friction, etc., then the entrance velocity R_1 relative to the vane would equal relative velocity r_1 at exit; the

absolute velocity v_1 of steam leaving the first set of moving vanes would equal the absolute velocity V_2 in passing through the intermediate set of guide vanes; and relative velocities R_2 and r_2 would be equal. The dotted arcs indicate which velocities are to be drawn equal in this construction.

Case Where There is Loss Through Friction.—If it is desired to take account of frictional loss this may be done as in Fig. 9, where relative velocity r_1 is made less than relative velocity R_1; absolute velocity V_2 less than v_1; and relative velocity r_2 less than R_2. The diminution of the velocities may be made either an arbitrary amount in each case or a certain percentage of the velocity, as desired.

It will be noted that in Figs. 8 and 9 the vane angles are different for the two wheels. If desired to make them the same for convenience in manufacture, a plan must be followed similar to that now to be described in connection with reaction blades.

The Vanes of Reaction Turbines.

In Fig. 10 is a diagram by which the action of the steam in reaction turbines may be studied. For convenience in manufacture the guide and moving vanes in any one step or series of the turbine are usually made alike. The upper parallelogram shows the absolute velocity and direction V of steam leaving the guide vanes, its velocity and direction R relative to the moving vanes at the point of entrance, and the velocity and direction w of the moving vanes.

The lower parallelogram shows the absolute velocity and direction v of the steam leaving the moving vanes, its velocity and direction r relative to the moving vanes at the point of exit, and the velocity and direction w of the moving vanes.

Characteristics of the Reaction Diagram.—The essential difference between this diagram and those for impulse turbines is that the relative velocity r is greater than R. In the reaction turbine steam first expands and acquires a velocity in the guide passages, as in the impulse turbine. Then, in flowing through the wheel passages it continues to expand and acquires a greater velocity *relative to the moving vanes* than it had at the entrance. The absolute velocity of the steam, of course, diminishes.

Call u the velocity acquired by the steam as a result of the expansion in the passages of any one row of vanes. Then, since steam enters each set of guide passages with an initial velocity equal to the velocity v of the steam leaving the wheel vanes, we have: Velocity V of steam leaving the guide vanes$=v+u$.

In the moving vanes the velocity u acquired through expansion has the effect of increasing the velocity of the steam relative to the vanes, so that relative velocity r of steam leaving the moving vanes$=R+u$. From the foregoing we therefore have, $V-v=r-R$, or the difference between the absolute velocities V and v equals the difference between the relative velocities r and R.

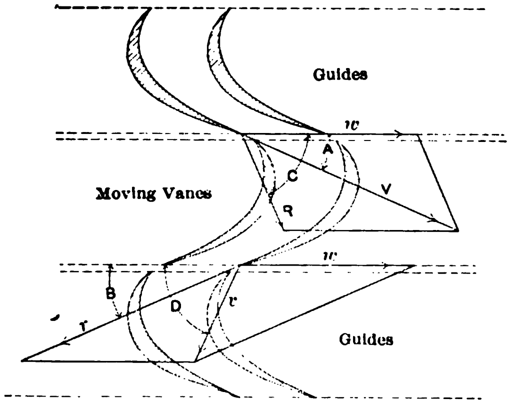


Fig. 10.

Construction of a Reaction Diagram.—In Fig. 10, where the guide and moving vanes are alike, it will be evident that angles A and B must be equal and angles C and D equal, to secure tangential action upon the vanes. Hence, in Fig 11 we make these angles equal, having A and B as small as practicable and making angles C and D from 70 to 90 degrees. In hydraulic work 90 degrees is frequently selected for this angle.

Referring to triangles (1) and (2), Fig. 11, we have the common side w and angles A and B equal; consequently these triangles are equal and $V=r$ and $R=v$. Having assumed a suitable value for w, and knowing the angles, the other sides may be calculated by the principle in trigonometry that in any triangle the

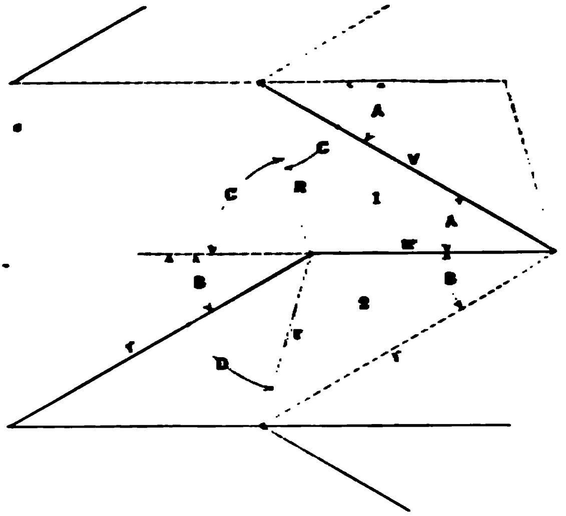


Fig. 11.

sides are proportional to the sines of the opposite angles, as follows:—

$$V=\frac{w \sin (180-C)}{\sin (C-A)}=\frac{w \sin C}{\sin (C-A)} \qquad (16)$$

$$R=\frac{w \sin A}{\sin (C-A)} \qquad (17)$$

$$v=\frac{w \sin B}{\sin (D-B)} \qquad (18)$$

$$r=\frac{w \sin D}{\sin (D-B)} \qquad (19)$$

The efficiency of the reaction turbine must be studied by taking the machine as a whole, since the action is continuous from the first to the last row of vanes and the losses through leakage, friction, etc., are such that no estimate of efficiency can be made by calculating the efficiency of any one set of guide and moving vanes. The efficiency can only be determined by experiment.

TESTS UPON BUCKETS AND CHANNELS.

Tests of Strickland L. Kneass, C. E.—In 1894 a small, experimental turbine was built and tested in the injector department of William Sellers & Co., Inc., Philadelphia. The machine was so designed that it could be assembled to operate on the plans of several types of turbines, including the Parsons, what later became known as the Curtis, and the De Laval; and when under the latter arrangement it could be run either as a compound or a simple turbine.

Preliminary to the turbine tests an investigation was undertaken of the action of steam jets upon vanes and in flowing through

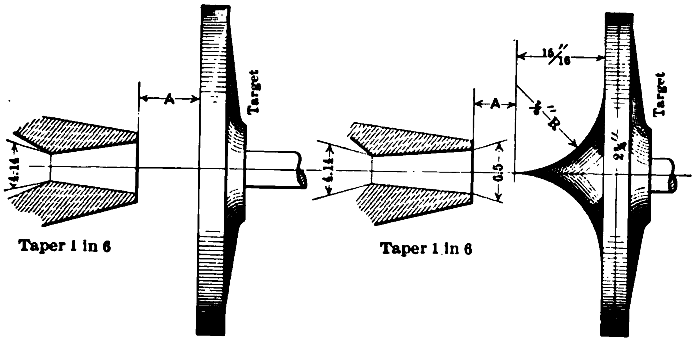


Fig. 12. Flat and Parabolic Targets for Measuring Impulse.

curved tubes. These, as well as the turbine tests, were at the inception of Mr. Kneass, whose experimental work has several times before been referred to, and who has allowed the author to make selections from the records of tests on buckets and tubes.

In the preliminary tests a delicately balanced target was employed, against which a steam jet from a nozzle was directed. The target was provided with a sensitive weighing device and was so manipulated in taking readings as to eliminate the effect of friction in the final results. In order to measure the impulse of the jet, the target was made with a parabolic surface coming to a point at the center, so as to deflect the stream through an angle of ninety degrees, with as little loss as possible. When vanes or passages of any particular shape were to be tested, the piece com-

prising these was bolted to the target, by which means the impulse or reaction could be measured.

Impact vs. Tangential Action.—In theoretical discussions of turbine vanes, it is assumed that the fluid must glide upon them tangentially in order to avoid losses from impact. In operation, however, hydraulic turbines seldom run at exactly the speed required to produce tangential action; and in fact, tests have not infrequently shown that the best results are obtained by running at slightly faster or slower speeds.

In elucidation of this subject are tests by Mr. Kneass, in which jets were allowed to discharge against flat and parabolic targets, as in Fig. 12, and the pressure measured in each case, with results as in Table I.

TABLE I.

Flat Target.			Parabolic Target.		
Steam Pressure, Gauge.	Distance A.	Pressure on Target per sq. Millimeter of Nozzle, Pounds.	Steam Pressure, Gauge.	Distance A.	Pressure on Target per sq. Millimeter of Nozzle, Pounds
120	1/8 inch	0.2302	120	1/8 inch	0.2413
120	1/4 inch	0.2320	90	1/8 inch	0.1777
120	1/2 inch	0.2390	60	1/8 inch	0.1109
120	1 inch	0.2461	30	1/8 inch	0.0540
120	1 1/2 inch	0.2461	...	......	......
Average,		0.2387			

These tests show, that by adjusting the distance of the nozzle from the target it was possible to secure as great a pressure with the flat as with the parabolic target and point to the conclusion that perfect tangential action of a jet upon a vane is not essential to high economy.

Another test illustrating the same fact was made with two different vanes, Fig. 13, one with an entrance angle of 60 degrees and one with an entrance angle of 20 degrees. The angle of the nozzle and the exit angle of the vanes were the same in both tests. The steam pressure was 15 pounds, gauge, and the pressures upon the target were, for the 60-degree angle

0.0454 pound and for the 20-degree angle 0.0475 pound, the jet flowing on tangentially at the latter angle. When the entrance angle was changed from 20 degrees to 60 degrees, there was a falling off of only 4.5 per cent.

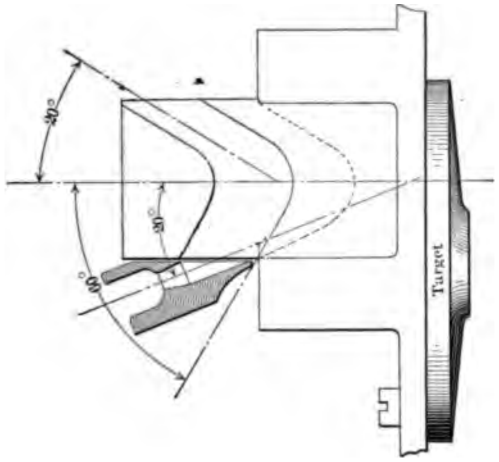


Fig. 13. Experiment with Different Entrance Angles.

Experimental Buckets.—Among the other buckets tested were those of Fig. 14 and selected results of tests upon them are given in Table II., which requires no explanation, since the reader can easily make his own comparison of data.

Owing to the falling off in the pressure on the target with bucket No. 6, as compared with bucket No. 1, another experiment was made with a bucket of similar design. The nozzle used with No. 6 was only 4 mm. diameter, while that used with No. 1 was 6 mm. diameter. In the additional experiment the nozzle was 7 mm. diameter, and beveled off as in the No. 6 test, but with a little larger passage leading to the nozzle proper. This gave a pressure against the target of 0.0572 pound per square millimeter of nozzle area, with an initial pressure of 15 pounds, gauge, showing a gain of 17 per cent over the experiment with the 4 mm. nozzle.

Experiments with Curved Tubes.—A series of experiments was made with curved tubing arranged as at *A* and *B* in Fig. 15

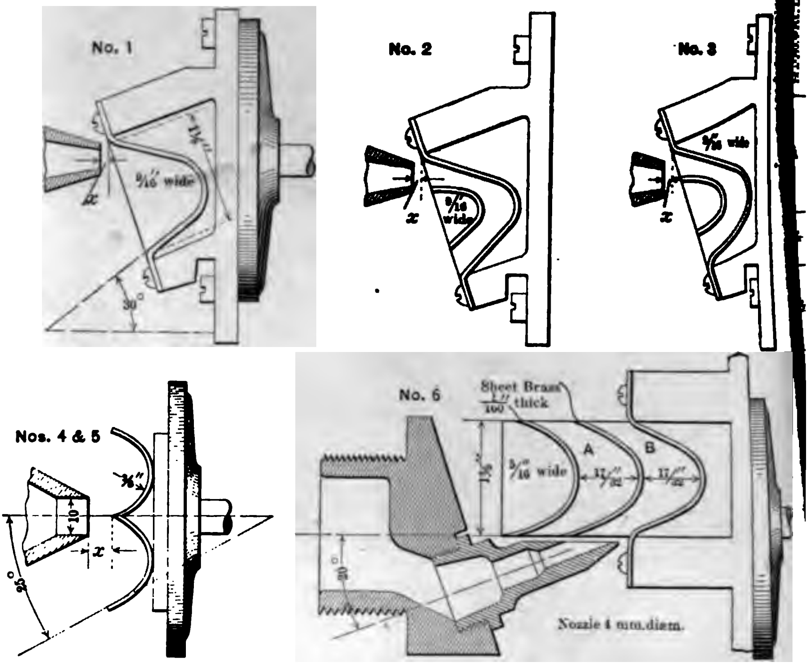


Fig. 14. Experimental Buckets.

which shows what large losses may result from friction. Copper tubing was used, 3/8 inch diameter, bent on an inner radius of 3/4 inch. The nozzle used was 1 in 6 taper, 4.14 mm. diameter at the throat and 7.5 mm. diameter at the mouth. Tests were first made with a single tube mounted on the target and bent through an angle of 180 degrees. Three similar bends were next used with the first two stationary and their ends separated 1/16 inch, as at *B*, and the third one mounted on the target. The results with this arrangement, when compared with the first set of results, show the losses in pressure due to friction in the tubes and disturbance caused by the spaces between the ends of the tubes. Finally, tests were made with a single tube mounted on the target, (this tube not shown) making 1½ turns, or the same number of turns all

TABLE II.

Number of Experiment.	Diameter of Nozzle in Millimeters.	Distance x in Inches.	Steam Pressure, Gauge.	Pressure on Target pr sq. Millimeter of Nozzle.	Remarks.
No. 1	6	$\frac{3}{16}$	12½	.0484	Received steam nicely into bucket, no backward discharge. At 120 pounds' pressure steam escaped in all directions from bucket. At other pressures there was little discharge from sides, and steam left in a good jet.
	6	$\frac{3}{16}$	28½	.0996	
	6	$\frac{3}{16}$	60	.1940	
	6	$\frac{3}{16}$	120	.3659	
	6	Close	15	.0607	
No. 2	6	$\frac{3}{16}$	12½	.0480	All steam entering channel.
No. 3	6	$\frac{3}{16}$	1:½	.0480	Apparently no resistance in striking knife edge.
	6	$\frac{3}{16}$	2-½	.0928	
	6	$\frac{3}{16}$	60	.1930	
No. 4	6	0	15	.0538	Bucket had open ends. Steam left bucket in a nice double jet. Knife edge was cut in the short time of the experiments.
	6	$\frac{3}{16}$	15	.0539	
	6	$\frac{13}{16}$	15	.0500	
No. 5	6	$\frac{3}{16}$	15¾	.0572	Same as No. 4, but ends of bucket were enclosed, thus confining the steam sideways.
	6	$\frac{1}{2}$	15¾	.0[illegible]27	
	6	$\frac{13}{16}$	15¾	.0620	
No. 6	4	Close	15	.0475	Discharged all into B.
	4	Close	15	.0419	Discharged equally into A and B.

told as the three tubes at *B* in the diagram. The pressures upon the target in the three cases, per square millimeter of nozzle area, are given in Table III.

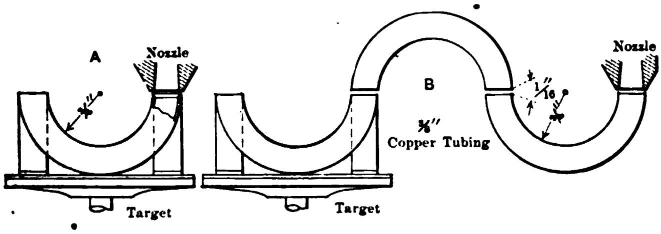


Fig. 15. Experiments with Curved Tubing.

Experiments with Rectangular Tubes.—Instead of continuing experiments with the round tubing, a curved rectangular tube was made with three half turns, as at *B*, Fig. 16. The dimensions of

Table III.

Pressures in Pounds per Square Millimeter of Nozzle, with $\frac{3}{8}$-Inch Tubing, Arranged as in Fig. 15.

Steam Pressure, Gauge.	Single Half Turn.	Three Half Turns—Pressure on Last One.	Continuous Coil, 1½ Turns.
120	0.400	0.248	0.371
90	0.248	0.170	0.260
60	0.174	0.098	0.159
30	0.084	0.043	0.066

the tube increased from 8.2 by 8.2 millimeters at the inlet to 8.75 by 12 millimeters at the outlet.

The nozzle used in this series was of 1 in 6 taper, 4.14 mm. diameter at the throat and 6.5 mm. at the mouth. It is the same nozzle used in the tests in Table I. with the parabolic target. Four arrangements were tried as illustrated at $A, B, C,$ and D in Fig. 16

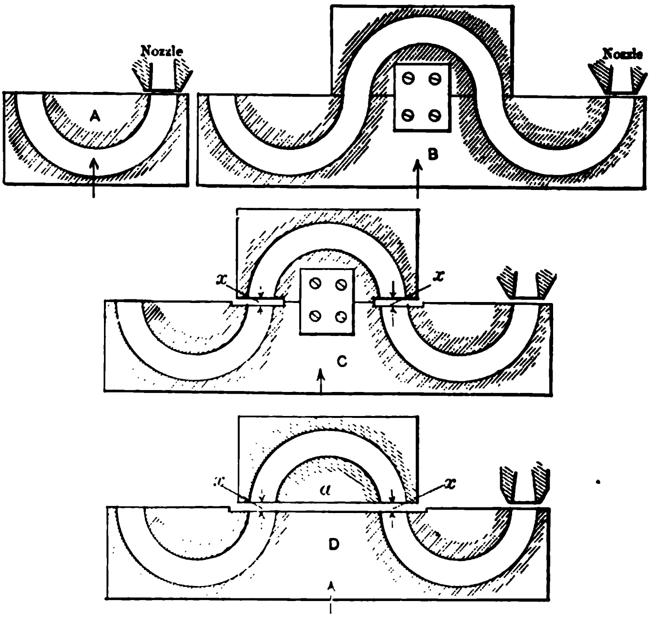


Fig. 16. Experiments with Rectangular Curved Tubing.

First, tests were made with a continuous bend, *B*, mounted on the target, with results tabulated in column 3, Table IV. Second, the three half-bends were cut apart, as at *C*, with openings $\frac{3}{64}$ inch wide, and all three bends located on the target, the results being tabulated in column 4 of the table. Third, a single half-turn, shown at *A*, was used, the results being given in column 2.

TABLE IV.

PRESSURES IN POUNDS PER SQUARE MILLIMETER OF NOZZLE, WITH $\frac{3}{8}$-INCH TUBING, ARRANGED AS IN FIG. 16.

Steam Pressure, Gauge.	Arrangement A.	Arrangement B.	Arrangement C	Arrangement D
120	0.4176	0.3720	0.3509	0.6805
90	0.3018	0.2659	0.2579	0.4994
60	0.1860	0.1632	0.1537	0.2998
30	0.0849	0.0742	0.0738	0.1414

Fourth, the arrangement at *D* was adopted, with the upper half-bend stationary and the two lower ones mounted on the target, results tabulated in column 5.

In comparing results it should be noted that, if there were no losses through friction, or otherwise, the pressures realized with arrangements *A, B,* and *C,* Fig 16, should be the same, since the tubes all turn the jet through 180 degrees at the point of discharge; and the pressures should also be double those obtained with the same nozzle discharging against the parabolic target, as given in Table I., where the jet is turned through an angle of only 90 degrees. With the arrangement at *D,* where the steam is twice turned through 180 degrees, and discharges twice at this angle, the pressures should be double those in the other cases with the tubes and four times those with the parabolic target.

Leakage of Steam.—In the foregoing tests where the tubing was cut apart there was a slight leak or overflow at the first opening, when initial pressure was 120 pounds, and much more at the second opening. At 90 and 60 pounds the leaking was much less, and at 30 pounds there was practically no leak whatever.

Another series of tests was run with a rectangular tube of slightly greater taper, hoping to have less leak at openings. Some

also made in the nozzle. With a continuous tube all red from the nozzle nicely; but when the three half- parated by $\frac{1}{32}$-inch spaces all the steam would not e was some waste at the mouth. There were also

TABLE V.

[illegible] CENT DROP IN PRESSURE, DUE TO OPENINGS IN TUBE.

St[illegible] Pre[illegible], Gauge.	[illegible]	[illegible]-Inch [illegible]enings.	$\frac{1}{8}$-Inch Openings.
121	6.[illegible]	7.5	13.6
90	3.8	4.8	10.0
60	2.0	2.0	9.3

leaks at each opening, but not as much as in the previous tests. At 90 pounds there was very little waste at the openings and none at the mouth. At 60 pounds a slight leak from second opening only and at 30 pounds no waste at all. The effect was then tried of increasing the width of openings from $\frac{1}{32}$ to $\frac{1}{8}$ inch and the pressures measured in each case. The readings were first taken with a continuous tube and then the tube having openings of $\frac{1}{32}$, $\frac{1}{16}$ and $\frac{1}{8}$ inch, the results being as in Table V.

CHAPTER XV

BODIES ROTATING AT HIGH SPEED.

Critical Speed of Rotating Bodies.—In the description of the De Laval turbine, Chapter III., reference was made to the so-called critical speed of the wheel and flexible shaft when rotating at high velocity. This phenomenon may be explained by the aid of the accompanying diagrams:

In Fig. 1 is a disk *W* mounted on a shaft *A B* turning in ball-and-socket bearings, as indicated. One side of this disk is supposed to have a dense section at *H*, making it heavier than the

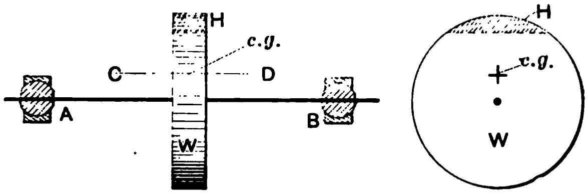


Fig. 1. Disk and Flexible Shaft.

opposite side. The center of gravity of the wheel, therefore, will lie to one side of the shaft *A B*, say on the axis *C D*. Now if this shaft and disk be rotated, the centrifugal force generated by the heavier side will be greater than that generated by the lighter side diametrically opposite to it, and the shaft will deflect toward the heavy side, as in Fig. 2, causing the center of the disk to describe a small circle, indicated by the dotted line at *a*. To locate the point at which a weight should be added, or on the other hand, at which metal should be drilled out in order to bring the piece into balance, a piece of chalk is held so that the high side of the disk will just touch it as it comes around. The weight necessary to balance, to be told by trial, is then added opposite to the high side where the mark appears; or else, if the balancing is to be done by drilling, metal is removed on the same side with the mark. In the most accurate balancing it is advisable to use a steel point held rigidly, but which can be fed up gradually until the point makes a faint scratch on the edge of the disk.

The foregoing conditions hold until a comparatively high speed is reached, depending upon the weight of the disk and flexibility of the shaft. A point will eventually be reached, however, at

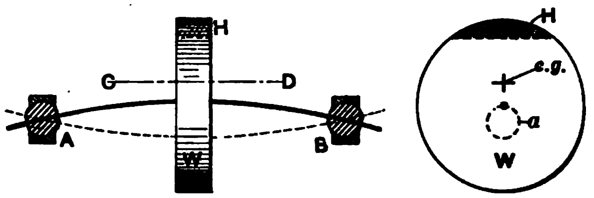


Fig. 2. Rotation about Geometrical Axis.

several thousand revolutions a minute, when there will momentarily be excessive vibration, and then the parts will run quietly again. The speed at which this occurs is called the critical speed of the wheel, and the phenomenon itself is called the settling of the wheel. The explanation is that at this speed the axis of rotation changes and the wheel and shaft, instead of rotating about their geometrical center, begin to rotate about an axis through

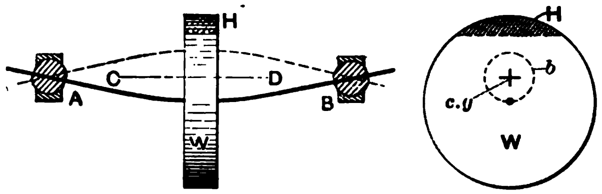


Fig. 3. Rotation about Axis of Gravity.

their center of gravity, or about the axis *C D* in Fig. 1. This is illustrated in Fig. 3, where the wheel and shaft have taken a new position in which the axis *C D,* if extended, would pass through the centers of the two bearings, while the shaft is deflected so that it traces a circle shown by the dotted line *b* in Fig. 3. It is to be noted, however, that this circle is now on the *H,* or heavy, side of the disk instead of on the other side as before, so that now if one were trying to locate the point where weight should be added in order to balance the disk, he would find that the chalk mark came on the light side of the disk, and that the weight should be added on the same side.

Point Where the Settling Occurs.—Mr. Konrad Anderson states that the settling of a rotating body occurs when the number of revolutions is equal to the number of vibrations which the shaft makes with the wheel mounted upon it. That is, a shaft and wheel have a certain time of vibration, just as does a pendulum or a spring, and when this synchronizes with the time of rotation the change is supposed to occur. In the De Laval turbine the flexible shaft and wheel are in such proportions that the settling takes place very quickly, and the critical speed is from 1/6 to 1/8 of the normal number of the revolutions of the wheel.

Disks Supported by a Rigid Shaft.—In the illustrations thus far shown, the body to be balanced is represented as a disk supported by a flexible shaft. While it is only in special cases that the flexible shaft would be used, it serves to illustrate the principle of balancing better than if the shaft were rigid. If a disk were mounted on a rigid shaft and the rotative method of balancing were to be applied, it would be necessary to support the shaft in bearings loosely connected to their pedestals, which would allow the shaft and disk to vibrate freely under the action of the forces generated.

The disks of the smaller Curtis turbines are balanced in this manner. The shaft is placed in a vertical position and suspended from the end of a cable which is given a rotary motion of the desired speed. The lower end of the shaft is steadied by a bearing held by springs, so that it is free to move in a sidewise direction under the influence of the rotating shaft. Both ends of the turbine shaft are thus flexibly supported and as the disk rotates the high side is marked with chalk or blue pencil held close to the rotating disk. The motion is then stopped, a small piece of wire twisted into the blades of the disk at the proper point for the purpose of bringing it into balance, and the operation repeated. When perfect balance is obtained, the pieces of wire show where metal must be added to or taken from the disk to complete the work.

Static Balancing may be successfully used for disks, if the apparatus is sufficiently delicate. Fig. 4 shows a machine used by the De Laval Steam Turbine Company for this purpose. The piece marked *A* is a coupling flange mounted on a vertical arbor

ready for balancing. The machine is placed under a drill press and by its aid the heavy side of the casting is located and enough metal drilled out to bring the flange into balance. On top of the stand are knife edges which carry a table *C* with a movable cross slide *B*. This cross slide is fitted with a pendulum in the form of

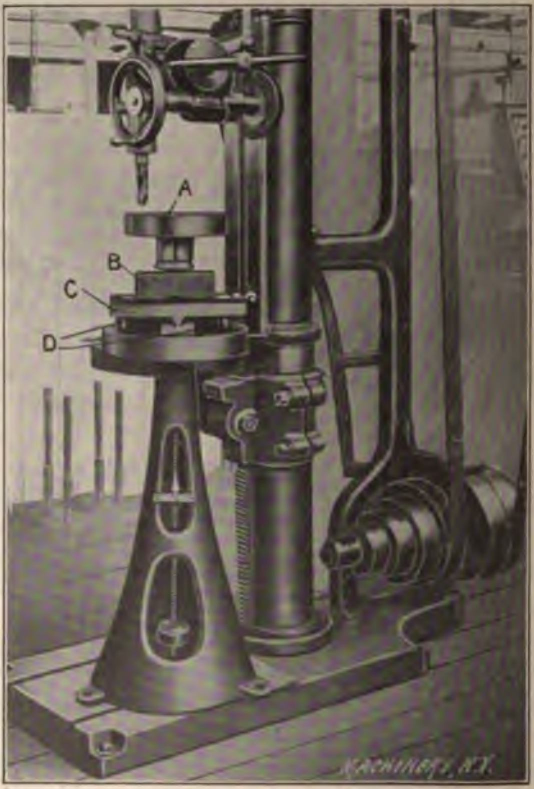


Fig. 4. Apparatus for Static Balancing.

a screw which runs down into the base and has a weight at its lower end. Half way up there is a pointer and a graduated scale for indicating the position of the pendulum. The arbor for supporting the piece to be balanced is at the top of the slide *B*. By adjusting the slide one way or the other the indicator is brought exactly at the center of the scale and then the coupling is turned

round half way by hand without moving the slide. If the indicator does not move, it shows that the coupling is in balance at this point; if it does move, the coupling must be out of balance and the necessary drilling is done. From eight to a dozen points around the circumference of the coupling are tested in this manner. In order to steady the table and avoid wear of the knife edges when drilling, there are pins D which are raised against the bottom of the table by a cam and thus take the strain.

Balancing Cylinders.—In attempting to balance a cylinder, like Fig. 5, a heavy portion might come at one end, as at H, and perhaps at the other end, also, but at a different point of the circumference, as at H^1, so that a considerable twisting moment would

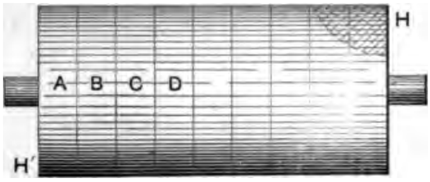


Fig. 5.

be introduced. It is desirable, if possible, to divide the cylinder into a number of disks, as A, B, C, D, and balance each one separately. But this cannot always be done, as in the case of turbine drums and the rotors of electric generators. The only way with such parts is to mount them in loose bearings supported by springs and then run them by motor or other means up to the required speed.

Locating the Heavy Side.—It is not always easy to tell by this means where the heavy spot is located, since, as explained in connection with the flexible shaft and critical speed, it may under certain conditions be on the same side with the high spot and under other conditions on the opposite side; and frequently, if the cylinder is approaching the critical point the heavy spot will lie somewhere between these two extremes. In the *American Machinist* for February 22, 1906, E. R. Douglas gives suggestions for finding the position of the heavy spot. After rotating the cylinder up to speed and marking the high spot on each end

with chalk, run it in the opposite direction and make similar marks. If they are in a different position from the first ones the heavy spot will lie half way between the two marks, but on which side can be told only by trial. Attach heavy balance weights at each end of the cylinder midway between the first and second marks. The weights should be heavy enough to completely outweigh the heavy spots. Now it will be evident that if the weights are in a position coinciding with the heavy spots, chalk marks made on the circumference as the cylinder is rotated will agree with the marks previously made when the cylinder was rotated in that direction; but if the weights are opposite the heavy spots, and are heavy enough to overpower the latter, then the new chalk marks will be opposite to the original ones, indicating that the balance weights should be attached in the position that the heavy ones now occupy.

Stresses in Rotating Bodies.

The calculation of the stresses in rotating bodies is often a complex problem. Fortunately the excellence of material now available makes well constructed turbine parts safe against bursting at the speeds that compound turbines usually run. In practice a factor of safety is needed, so that approximate methods may be used in calculations, if the factor is on the safe side. If we are dealing with a disk in which there are both tangential and radial stresses (Fig. 6), we might neglect the considerable effect of the radial tension and suppose the disk to be made up of a series of concentric rings, in which the tangential or hoop tension only would be considered. Such a method, although only roughly approximate, would be safe, since there would actually be the additional radial tension to help. This method would be preferable to calculating the *average* tensile stress across the whole cross section of a disk, as is sometimes done, since the strength of a disk is at its weakest part and it would give way at the point where the tension was the greatest. (See Weisbach's *Theoretical Mechanics*, page 618, for method of calculating the average stress.)

Stresses in a Rotating Ring.—When a cylindrical ring that is comparatively thin radially, like the rim of a flywheel, is rotated

about its axis, tension is set up proportional to the weight of the material and the square of the circumferential velocity. This tension is due to centrifugal force.

The formula for the stress is

$$f = \frac{12\,wv^2}{g}$$

where f=stress in ring, pounds per square inch.
w=weight of 1 cubic inch of material.
v=velocity of ring in feet per second.
g=32.2.

Example.—At a speed of 400 feet per second what tension would be set up in a ring, one cubic inch of which weighs 0.28 pound?

$$f = \frac{12 \times 0.28 \times 160{,}000}{32.2}$$

=16,700 pounds per square inch, or about the bursting strength of cast iron.

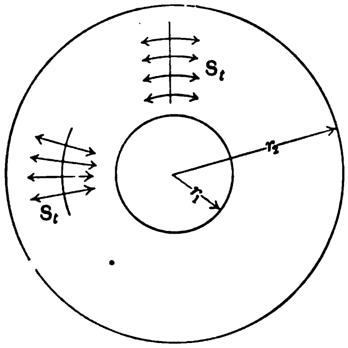


Fig. 6.

*Stresses in a Rotating Disk.**—An exact analysis of this problem is given by Stodola in his work on steam turbines. In the general case in which the disk is of varying width, the equations giving the stresses are complicated, but for the case of uniform

*Prepared at the request of the author by Prof. G. A. Goodenough, of the University of Illinois.

thickness the equations reduce to comparatively simple forms. Without any attempt at derivation we give the final results.

Let w=weight of material per cubic inch.

N=speed, R. P. M.

r_1=inner radius in inches.

r_2=outer radius in inches.

k=a constant (Poisson's ratio) usually taken as 0.3 for iron or steel.

r=radius of any point at which stress is desired.

S_t=tangential stress, lb. per sq. inch.

S_r=radial stress (See Fig.)

Then

$$S_t=0.00000355\,w\,N^2\left[(3+k)\left(r_1^2+r_2^2+\frac{r_1^2r_2^2}{r^2}\right)-(1+3k)r^2\right].$$

$$S_r=0.00000355\,w\,N^2\,(3+k)\left(r_1^2+r_2^2-\frac{r_1^2r_2^2}{r^2}-r^2\right).$$

For $r=r_2$, that is, at the outer circumference,

$$S_t=0.00000355\,[(3+k)\,(2r_1^2+r_2^2)-(1+3k)r_2^2].$$

while at the inner circumference where $r=r_1$,

$$S_t=0.00000355\,[(3+k)\,(r_1^2+2r_2^2)-(1+3k)\,r_1^2].$$

It will be observed that the thickness of the disk has no influence on the stress. The factor outside the brackets depends only in the material of the disk and the speed: that inside the brackets upon the dimensions of the wheel. Evidently the stresses increase as the square of the speed.

Example.—Take a flat disk 40 inches outside and 4 inches inside diameter, material weighing 0.28 pound per cubic inch and running at 1,000 R. P. M. Taking k=0.3 the tensile stress at the inner circumference is

$$.00000355\times0.28\times(1{,}000)^2\,[3.3\,(2\times20^2+2^2)-1.9\times2^2]$$
$$=2{,}600 \text{ pounds per sq. in. approx.,}$$

and at the outer circumference it is

$$.00000355\times0.28\times(1{,}000)^2\,[3.3\,(20^2+2\times2^2)-1.9\times20^2]$$
$$=580 \text{ pounds per sq. in. nearly.}$$

Of course, the larger stress is taken as determining the strength of the disk. From the form of the general equation it is seen that the maximum tangential stress will always be at the inner circumference.

The circumferential speed in the example above is about 175 feet per second. If the disk were run at five times this speed, which is something like that of a De Laval turbine disk of this diameter, the stresses will be twenty-five times as large—65,000 pounds per square inch for the inner circumference and 14,500 pounds per square inch at the outer circumference. In the De Laval turbine, however, the disk is not made of uniform thickness, and the design is such that the tangential stresses are the same at all radii.

CHAPTER XVI

NOTES ON EFFICIENCY AND DESIGN.

Efficiency of a Turbine.—On page 181 is an explanation of the thermal efficiency of turbines, useful in comparing the performance of different turbines. In estimating the losses in a turbine another method for computing efficiency is used, which gives entirely different and much higher results. It consists in calculating the rate of steam consumption for an ideal turbine, in which there are assumed to be no losses of any kind, and then finding the ratio of this to the rate of steam consumption for an actual turbine.

In an ideal turbine steam would expand adiabatically from the initial to the final pressure and the energy can be calculated by any one of formulas (2), (7) or (8) for saturated steam; or (11) and (12) for superheated steam, Chapter XIII.

For example, in (8) the foot pounds of energy per pound of steam are represented by

$$J[\lambda_1 - T_2(\phi_1 - \theta_2) - q_2] \qquad (1)$$

One horse-power is equivalent to 33,000 foot pounds per minute, or $33,000 \times 60$ foot pounds per hour. Hence, the number of pounds of steam per horse-power per hour required by the ideal turbine, in which steam expands adiabatically, is

$$\frac{33000 \times 60}{J[\lambda_1 - T_2(\phi_1 - \theta_2) - q_2]}, \text{ or}$$

$$\frac{2544.98}{\lambda_1 - T_2(\phi_1 - \theta_2) - q_2} \qquad (2)$$

Example.—Steam expands adiabatically in a turbine from 165 pounds absolute to 1 pound absolute. The rate of steam consumption is

$$\frac{2,544.98}{1,193.6 - 562.69(1.5581 - 0.1329) - 70}$$

which reduces to the fraction

$$\frac{2{,}544.98}{321.65} = 7.912 \text{ pounds.}$$

Now suppose an actual turbine to consume 12 pounds of steam per brake horse-power per hour at normal load. Its efficiency would be $7.912 \div 12 = 0.659 = 65.9$ per cent.

Analyzing Turbine Losses.—By the foregoing method the total losses in a turbine can be determined; but it is a difficult matter to subdivide these, finding what per cent is due to one cause and what to another.

It has been shown by H. F. Schmidt* that a step in this direction can be taken for a given turbine, by plotting the total steam consumption curve for the turbine. It is well known in engine practice that this curve, when plotted for a throttling engine, becomes a straight line, while for an automatic engine it is usually a curved line. In the case of the turbine, it is approximately a straight line, without regard to type.

In Fig. 1 the ordinates of the chart represent pounds of steam per hour and the abscissas electrical horse-power. Points were plotted for the weight of steam used per hour by a 500 Kw. turbine at different loads and the line *AD* drawn through them, which is the total steam consumption line for the turbine. This line cuts the vertical coördinate at a point above the base line and intersects the latter to the left of the zero point.

A similar line *C*0 was then drawn for an ideal turbine operating without losses between the same pressures as the actual turbine. Since there are assumed to be no losses, the weight of steam used per hour is directly proportional to the load and the line for the ideal turbine passes through the zero point of the chart.

Now, it will be evident that a certain amount of power is required to drive the rotor of a turbine, and overcome the journal friction, the windage of the generator fields, and the friction of the drum or disks rotating in the atmosphere of steam encased in the turbine. The loss due to these causes is constant, or nearly

*Notes on the Steam Turbine, *Street Railway Journal*, June 25, 1904.

so, at all loads, while another class of losses, such as leakage, nozzle losses, etc., is directly proportional to the load. At no load, or where the power delivered by the turbine becomes zero, the variable losses disappear while the constant losses become the load of the turbine.*

Hence $0D$ on the diagram represents the power absorbed by the constant losses at zero load, and therefore at all loads. $B0$, drawn

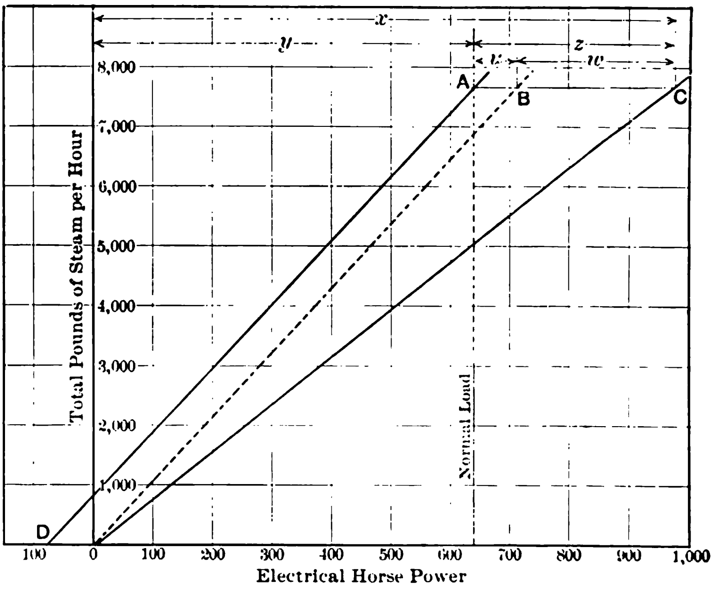


Fig. 1. Diagram of Total Steam Consumption Curves.

through 0, parallel with AD, is the total steam consumption line for the turbine, with the constant losses deducted, and the inclination of $C0$ to $B0$ shows how the variable losses increase with the load.

The power represented by the losses in the turbine can be estimated from the diagram. Suppose we wish to find what it is at

*As a matter of fact the losses due to leakage, friction in the nozzles, etc., do not entirely disappear at no load, when a turbine is running light, but they are very small in amount and in our analysis must be classed with the constant losses.

640 horse-power. From this point on the base line, Fig. 1, trace vertically to A, and then horizontally to C. Distance x shows the power that could theoretically be developed by the steam if there were no losses; y the part of this power that goes into useful work; z the part that is wasted; w the part absorbed by the variable losses; and u the part absorbed by the constant losses.

Variable Losses.—Having separated the constant from the variable losses, the latter can be divided into their elements only by a study of the conditions and of such tests upon nozzles, vanes, etc., as are available.

The variable losses are due chiefly to:

1. Leakage.
2. Radiation.
3. Residual velocity of the steam leaving the last row of buckets.
4. Imperfect action of the steam in the nozzles and blade channels, due to friction, eddying, etc.

The latter (No. 4) may be placed under two heads, one called "thermal" and the other "hydraulic." The thermal losses are caused by the failure of the nozzles and blade channels to properly expand the steam and convert its heat energy into mechanical energy, while the hydraulic losses are caused by the failure of the moving vanes to convert all the mechanical energy of the steam into work at the spindle.

Losses in 1,250 Kw. Turbine.—To illustrate what has preceded, take the second test on the 1,250 Kw. Westinghouse-Parsons turbine, page 190. This test was with saturated steam, with a mean initial pressure of 143 pounds and a condenser pressure of one pound absolute.

The theoretical steam rate, calculated by formula (1) is 7.928 pounds per horse-power hour.

The results of the test are given in terms of electrical horse-power, and it will simplify the analysis to eliminate the generator efficiency from the results and plot the diagram in terms of brake horse-power. On page 178 the efficiencies of the generator of this turbine are given as follows: Full load, 0.96; half load, 0.93; quarter load, 0.86, and by using these the brake horse-

able energy in the steam. Hence, power that could theore be produced $=1,904.5 \div 0.589 = 3,233.4$; and $3,233.4 - 1,9$ $1,328.9 =$ power required to overcome losses. Of this, 275 power (from the diagram) is required for the constant and $1,328.9 - 275 = 1,053.9$ for the variable losses at full lo

Since the efficiency of this turbine is 58.9 per cent the amount to 41.1 per cent; and of these, 275 horse-power, o per cent (obtained from the chart), are required to turn the leaving 26.7 per cent for distribution among the several iten make up the variable losses.

Unfortunately, there are no available data by which the sis can be continued in the case of the Parsons turbine. not know the residual velocity, and it can only be determin the application of a gauge or manometer to the last step turbine, by which the pressure can be found. This velocity assumed and the radiation assumed, leaving the losses from age and imperfect action in the blade channels to be obtain subtraction; but the results would be but little better than a in the absence of definite data.

Notes on Design.

In turbines which are divided into stages, it is desirable to an equal amount of energy utilized in each stage, and an i tant problem is to determine the steam pressures and qual steam in the different stages under these conditions.

If expansion were adiabatic and the water of condensation to settle out of the steam in each stage, leaving the stean the number of stages and the pressure at each stage could be approximately by aid of the charts at the end of the vo Thus, if a turbine is to operate between steam pressures o pounds and one pound absolute and the steam is to have a ve of 1,200 feet per second at each stage, we find from diagrams 2 that the pressure will drop in the first stage to 117 poun the second to 82 pounds; in the third to 57 pounds, etc., ap mately 13 stages being required in all.

This assumption in regard to the dryness of the steam, how is probably not in accord with the conditions and the proble

be more satisfactorily handled by aid of the temperature-entropy diagram.

Temperature-Entropy Diagram for Stage Turbine.—Assume the initial and final pressures to be 165 pounds and one pound, respectively, the turbine to have three stages and the steam to expand adiabatically. By formula (8), Chapter XIII., the kinetic energy acquired per pound of steam is found to be 250,251 foot pounds, requiring an expenditure of 321.66 heat units. For the construction of the temperature-entropy diagram we have the following data:

Temperature saturated steam at 165 lb. abs. = 365.88.
Entropy of water at 165 lb. abs. = 0.523.
Entropy of steam at 165 lb. abs. = 1.558.
Temperature saturated steam at 1 lb. abs. = 101.99.
Entropy dry saturated steam at 1 lb. abs. = 1.985.

Points *b* and *c*, Fig. 3, represent the entropy of water and steam respectively for the higher temperature. Through *b* draw the straight line *ba* intersecting the 32-degree point on the vertical coördinate. This is the water line of the diagram and closely approximates the true water line, which is a logarithmic curve. The entropy of point *a* at a temperature 101.99 degrees is found by measurement to be 0.11. Through *a* draw the horizontal line *ad*, and through *c* the adiabatic line *cd*. The line *ce* is the saturated steam line and the entropy of point *e* = 1.985.

The area *abcd* represents the number of available heat units per pound of steam and should equal 321.66, but actually equals 327.61, because the water line was taken as a straight instead of a curved line.

We now have the geometrical figure *abcd* in the form of a trapezoid, to be divided into three equal parts, each part representing the energy expended in one stage of the turbine. This is a geometrical problem, merely, and may be solved by the following formula, taking the values in terms of temperature and entropy units.*

Let A = length of top of trapezoid in entropy units.
B = length of bottom in entropy units.

*Formula proposed by Ralph E. Flanders.

C=height in degrees F.

P=percentage of whole area which is to be included between a line horizontal with the base and the base itself.

H=height of horizontal line above base in degrees F.

Then:

$$H=\frac{C}{B-A}\left(B-\sqrt{B^2-P(B^2-A^2)}\right) \qquad (3)$$

In the example above $A=1.558-0.523=1.035$; $B=1.558-0.11=1.448$; $C=365.88-101.99=263.89$; $P=\frac{1}{3}$ and $\frac{2}{3}$. Substituting in (3) and solving, $H=78.59$, when $P=\frac{1}{3}$, and 165.49 when $P=\frac{2}{3}$.

Taking these values of H, the horizontal lines fg and hk are drawn, dividing the diagram into three equal areas. The temperature represented by hk is $101.99+78.59=180.58$; and by fg is $101.99+165.49=267.48$.

In Fig. 3 ce is the saturated steam line of the diagram and the amount of dry steam present at the end of adiabatic expansion is

$$\frac{ad}{ae}.$$

Diagram Showing Reëvaporation.—Under actual conditions there would be friction and eddying of the steam which would retard the velocity of flow and part of its kinetic energy would be converted into heat, which would reëvaporate some of the moisture, making the steam drier than indicated by the above ratio.

Fig. 4 shows how this action may be represented by the heat diagram. Taking the same pressures as before, assume that steam flows from the higher to the lower pressure and that, after discharging, its velocity is checked by eddying or otherwise, so that 20 per cent of its kinetic energy is converted into heat energy.

The available heat energy is represented by area $abcd$, containing 327.61 heat units as in Fig. 3.

We will assume two different conditions of discharge.

First, that it is into a space so large or unconfined that the reëvaporation will have no tendency to raise the steam pressure.

The evaporation of the moisture will then take place at constant temperature and the change in entropy will be along the isothermal line *de*. If this change is *dh*, the area *d'dhk'* under *dh* will represent the work of reëvaporation, or $327.61 \div 5 = 65.52$ heat units, the reëvaporation being 20 per cent.

The line *dh* is at an absolute temperature of $101.99 + 460.7 = 562.67$ degrees and *dh* will therefore have a value of $65.52 \div 562.69 = 0.116$ entropy unit. This added to 1.558, the entropy of point *d*, gives 1.674 as the entropy of the steam at the completion of the reëvaporation. The dry steam present is therefore

$$\frac{ah}{ae} = \frac{1.674}{1.985} = 0.84, \text{ or } 84 \text{ per cent.}$$

In the diagram the area *afgd* is equivalent to the area *d'dhk'*, which, deducted from *abcd* leaves *fbcg* as the net heat energy converted into useful work.

Second: Another way to regard the matter is to assume the steam to discharge from the nozzle into a closely confined space so that the reëvaporation will raise the pressure and temperature of the steam. If this change is adiabatic and 20 per cent of the energy is expended in reëvaporation, it is obvious that the temperature will rise until it reaches a point *g*, where the percentage of dry steam is 84, as before determined. To find the quality of the steam, first locate *fg*, cutting off 20 per cent from the total area of the diagram. By formula (3) it is found to be 46.64 degrees above *ad*, giving a temperature of $101.99 + 46.64 = 148.63$ degrees. Entropy of dry, saturated steam at this temperature (point *l*, Fig. 4) $= 1.841$; entropy of steam discharging from nozzle remains constant at 1.558; dryness of steam $= 1.558 \div 1.841 = 0.84$, as found by the other method.

A curved line drawn from *c* to *h* will give approximately the quality of the steam at any point.

Fig. 5 is a reproduction of Fig. 3, but with the dotted lines *f'g'*, *h'k'* and *a'd'*, showing the net work and the temperature of each stage under the assumption that expansion is adiabatic, reëvaporation is 20 per cent and that the latter action is adiabatic, as above explained. Then:

Net work in first stage is $f'bcg'$ and temperature is x' instead of x; net work in second stage is $h'f'g'k'$ and temperature is y' instead of y; net work in last stage is $a'h'k'd'$ and final tempera-

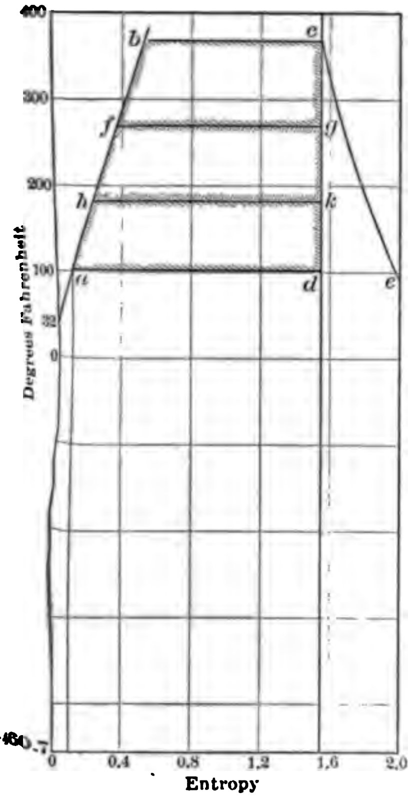


Fig. 3. Temperature-Entropy Diagram for Stage Turbine.

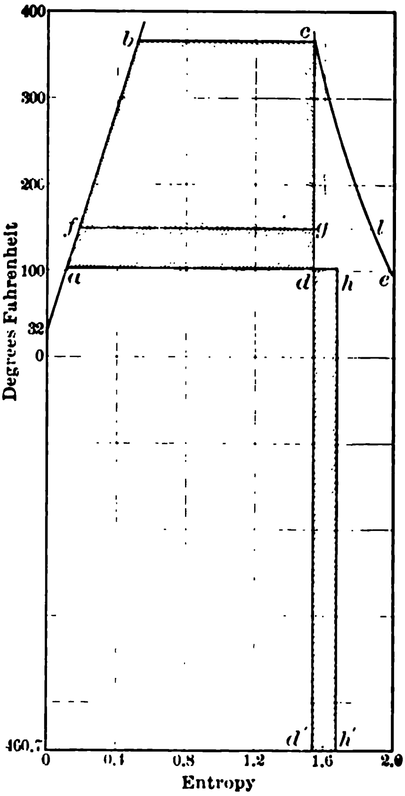


Fig. 4. Temperature-Entropy Diagram Showing Re-evaporation.

ture is z' instead of z, showing that the heat given up by friction and eddying is carried along from stage to stage, and discharged to condenser at the end.

The temperatures x', y', and z' enable the quality of the steam to be found in each stage by which the area of the passages may

be calculated; and also the pressure differences existing, by which the velocity of flow may be calculated.

Example in Design.—Required, to proportion a 500 Kw. turbine, multicellular type, with one wheel in each compartment; pressures 165 pounds and one pound absolute.

Tests on a 500 horse-power Rateau turbine of this type show a brake efficiency of 60 per cent. Thirteen per cent of the power at normal load is required to turn the rotor, leaving 27 per cent to be distributed among the other losses.

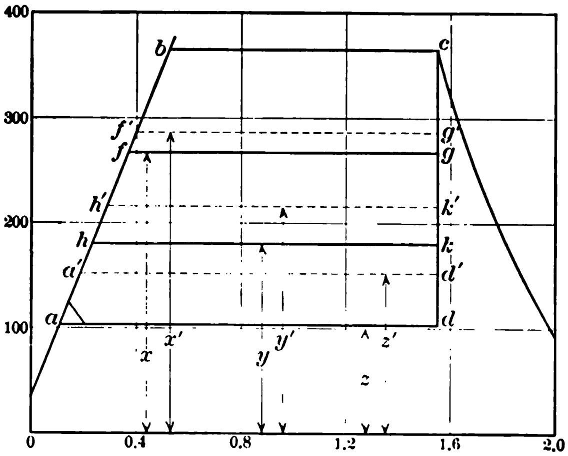


Fig. 5. Diagram Showing Re-evaporation.

The turbine will be governed by throttling the steam and we will assume that the initial pressure is throttled to 100 pounds absolute at normal load, leaving a small margin for overloads without resort to the secondary admission valve.

There will be 10 stages, this being sufficient to produce a moderate wheel velocity and to ensure that the drop in pressure from stage to stage will not exceed 40 per cent, so that diverging nozzles will not be necessary.

The energy available is found by formula (1) to be 289.36 heat units, or 225,122 foot pounds; and 22,512 per stage, giving a velocity of discharge of 1,204 feet per second. Tests with converging nozzles indicate that the actual velocity will be within

2 per cent of the calculated; and the actual velocity of discharge may therefore be taken at 1,180 feet per second.

Making the wheel vanes symmetrical, we find, formula (10), Chapter XIV., wheel velocity $w = 628$ feet per second; and by formula (11) residual velocity of steam $= 430$ feet per second.

To determine the area of the guide passages or nozzles three additional items must be known: (1) the weight of steam required; (2) the pressure at each stage; (3) the dryness of the steam at each stage.

1. With a turbine efficiency of 0.60 and a generator efficiency of 0.94, the combined efficiency will be 0.564. To obtain weight of steam, 500 Kw. $= 670$ H. P. and $(670 \times 500) \div 0.564 = 653,$-369 foot pounds that must be provided for per second. The available energy per pound of steam was found to be 225,122 foot pounds. Hence, weight of steam required $= 653{,}369 \div 225{,}122 =$ 2.9 pounds per second.

2. The pressure in each stage is to be determined by the aid of formula (3), following the approximation of either Fig. 3 or Fig. 5, and carrying through the calculation for the 10 stages. This will give the temperatures for each stage, from which the pressures can be obtained.

3. The dryness of the steam at the different stages is to be obtained from the diagram by estimating the evaporation as already explained; either doing this at the end of expansion and sketching in the temperature-entropy line for the mixture of the steam and vapor (*ch* in Fig. 4), or by calculating the entropy at different points and plotting the curve.

To arrive at the probable reëvaporation, let us assume the losses as follows:

		Per cent.
1.	Constant (to turn rotor)	13
2.	Radiation and leakage	5
3.	Residual velocity (last stage)	2
4.	Friction in nozzles	2
5.	Friction and eddying in channels, etc	18
		40

Of these, a part of (1), due to friction of the rotating disks in the steam, represents conversion of mechanical work into heat. If the turbine is designed so that steam blows directly from one stage into the next, only a part of (5) will cause reëvaporation; but if the steam is brought to a standstill in each stage, the whole 18 per cent will act in this way. It probably will not be far wrong to take the reëvaporation at 20 per cent.

It will be realized that calculations based on normal load become null and void at other loads and that the turbine problem is one of compromise between different running conditions. The final and most important determination of proportions must be empirical, based on tests of the completed machine.

CHAPTER XVII

THE COMMERCIAL ASPECT OF THE TURBINE.

Limitations of the Turbine.—The field of the turbine is limited by its relatively high speed and the facts that it is a one-speed machine and is non-reversible, in the sense that the reciprocating engine is reversible, without duplication of its main running parts. The high speed of the turbine precludes its use for driving machinery by belt or ropes, unless some form of balanced reduction gearing is employed, which no mechanic would advocate, except for the smallest units. Because of these limitations the turbine is restricted to driving direct-connected apparatus, such as generators, centrifugal pumps, blowers, and the propellers of ships; and in the latter case special expedients must be adopted to secure the necessary speed variation and reversal, as explained in Chapter XX. The turbine is essentially a central station machine, for the generation of electric power.

The Field of the Turbine.—The extent to which the turbine has laid hold of the central station field may not be generally appreciated, but is shown in a striking manner by the records of sales of the turbine manufacturers. Up to April, 1906, as recorded in the National Electric Light Association Report upon steam turbines for that year, the Westinghouse Machine Company had either installed or on order turbines for 180 customers; and of these 120 were for electric light, traction or power companies. Also, in a list of turbines of 500 Kw. or over, installed by the General Electric Company, or ordered of them, about 140 out of 175 purchasers were electric light, traction or power companies.

A similar condition also exists in England and on the continent. In the paper just mentioned, is a report by W. C. L. Eglin, who traveled abroad to investigate the turbine situation. He says: "There can be no doubt as to the status of the turbine as a prime mover, in the generation of electrical energy by the use of steam, for all of the recently designed stations which were visited were equipped with turbines and some of the older stations, built originally for engine-driven units, are being increased in capacity by installation of turbine units. The only observed instance in

which this is not true is that of the Metropolitan station in Paris, which is being completed by the installation of one large engine-driven unit. . . . . It is rather interesting to note that in interviews with leading engineers no question was ever raised as to the comparative merits of engines and turbines for electric light plants."

While the turbine in all sizes is very successful for electric generating, the distinct field that it has won for itself is that of large units, in central stations. Large steam engines, with their heavy frames, ponderous moving parts and large generators, are in marked contrast to the small and compact turbine units of corresponding power.

The Field of the Reciprocating Engine.—The power for rolling mills, blast furnaces, waterworks, mine hoisting, air and ammonia compressing, etc., will be furnished by the piston engine for a long time to come. Rolling mills have been driven by electricity to a limited extent, and centrifugal air compressors, turbine driven, have been successfully used. But in general the turbine must wait upon the development of such apparatus before it can enter the above fields. It is also safe to say that the Corliss or similar type of engine will continue to be used for mill work, where driving by belt or ropes is in vogue.

The competition to be met in electric generating will depend upon the future development of the turbine. The design of Parsons turbines is not well adapted to small-sized units, and turbines of this type are not now built in this country in powers of less than 400 Kw., or about 600 horse-power. As long as these turbines are not made in the smaller sizes, the only competition with reciprocating engines of less than 600 horse-power will be turbines of the impulse type, such as the De Laval and the Curtis, and there is thus a comparatively clear field for the several types of engines made in these sizes. Their real competitor at the present time is the gas engine rather than the turbine.

The possibility of engines of intermediate sizes, say from 600 to 1,500 horse-power, meeting turbines on an equal footing in competition depends mainly upon whether land values and space available are at a premium. If such is the case the turbine would naturally be selected; but if not, many engineers would select in

preference compound engines of medium speed, which are economical and reliable and reasonably compact.

Turbine Advantages, as usually claimed, are about as follows: High economy under variable loads; small floor space; uniform angular velocity and close speed regulation; freedom from vibration; inexpensive foundations; ease of erection and quickness in starting; steam economy not seriously impaired by wear or lack of adjustment; small cost for maintenance and attendance; but little danger from water entrained in the steam; adapted for high superheat; water of condensation free from oil.

Reciprocating Engine Advantages.—Rather than call attention to special features, engine builders point to the proven reliability of the reciprocating engine; to the fact that it is in no sense an experimental or undeveloped device; that its condensing system is simple, requiring only a small quantity of cooling water; and that high economy is obtained without the use of superheated steam.

Several of these claims for the turbine and engine will bear discussion.

Comparative Economy.—This has been considered in its different phases in Chapters IX. and X. A point in this connection, not as generally appreciated as it should be, is that a compound engine will not show up creditably under variable loads unless properly designed and adjusted. It is held by some authorities that under variable loads the best results cannot be obtained with the drop cut-off gear of a Corliss engine. With this gear the initial pressure in the cylinder approximates the steam pipe pressure at all points of cut-off, and it is held that better economy is to be obtained by throttling the steam in the high-pressure cylinder at short cut-offs. This is easily accomplished by the use of shifting eccentrics and a shaft governor.

In the matter of adjustment, tests on one of the 7,500 horsepower engines of the 59th Street station of the Interborough Rapid Transit Company, New York, show what effect this may have. When running with a load equally divided between two cylinders the steam rate at 4,000 Kw. was 17.4 pounds and at 6,000 Kw. 19 pounds. Afterwards, by adjusting the low-pressure gear the receiver pressure was changed, with results at these two

loads of 17 and 17.5 pounds per Kw. hour.* It is reasonable to suppose that a large proportion of compound engines, even if designed along the lines of best economy, are not running under proper adjustment, and this fact must be considered in connection with the comparative economy of turbines and engines; for there are no adjustments to be made on the turbine that can seriously affect its rate of steam consumption.

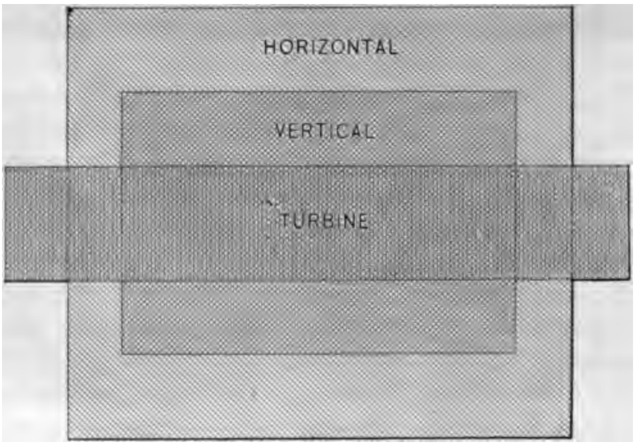


Fig. 1. Relative Floor Space for 1,000 Kw. Engines and Turbines.

Electric Generating.—In running direct-connected alternators in parallel, the turbine has the advantage of a uniform turning moment and its high speed produces a powerful regulating force without the use of a flywheel. There is no reciprocating motion to be converted into synchronous rotary motion. The regulation of turbines is so close that it has been found possible to run railway, power and lighting circuits from the same machine. Where a turbine is installed in a plant with piston engines or water wheels, it tends to have a steadying influence on the whole system, owing to its inertia effect.

The Use of Oil.—No cylinder oil is required for the turbine, so that the exhaust may be condensed and used over and over again in the boilers, provided precautions are taken to prevent

*See diagram in paper by Henry G. Stott on Power Plant Economics, Proc. Am. Inst. E. E., January, 1906.

oil coming from the condenser auxiliaries intermingling with the condensed steam. The bearings of the turbine are the only parts that are oiled, and as the lubricant is circulated through the bearings, then collected, cooled and used over again, there is but little opportunity for loss. Returns from a large number of users of Westinghouse turbines show that only about one quarter gallon of oil is required for the bearings per kilowatt per year, which is at the rate of about five cents a day for a 400 Kw. turbine.

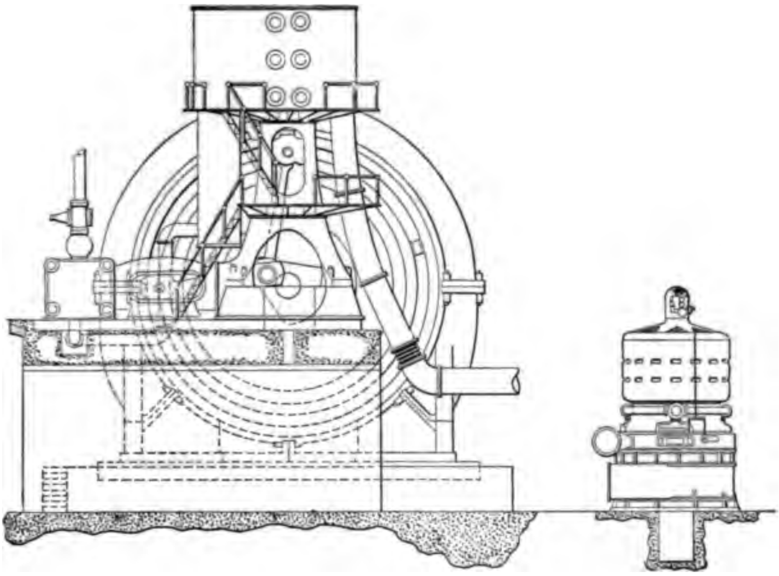


Fig. 2. Comparison of 5,000 Kw. Units.

Noise.—There is a great difference in the running qualities of different turbine generators, some of them operating quietly and others producing a roar that is very objectionable. This has been overcome to some extent by designing the rotating fields so as to have a smooth exterior, and more recently, in the Parsons type of turbine, the noise has been largely obviated by encasing the generator and then circulating air through the casing by means of a blower. The casing deadens the noise and the circulation of the air cools the motor and enables heavier overloads to be carried for longer periods of time.

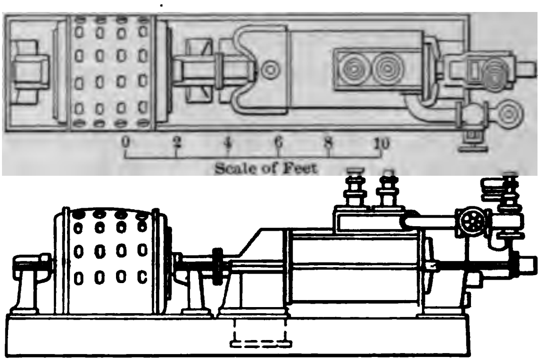


Fig. 3. Plan and Elevation of 500 Kw. Westinghouse Turbine.

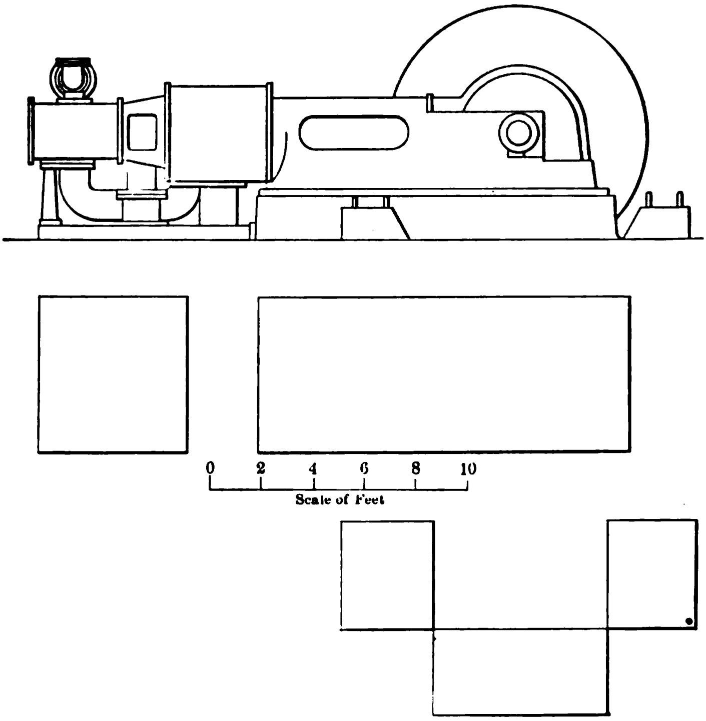


Fig. 4. Elevation and Plan of 500 Kw. High-Speed Engine.

Relative Space Occupied by Engines and Turbines.

Comparison of 500 Kw. Units.—Fig. 1 is a graphic comparison of the floor space required for horizontal turbines, and vertical and horizontal cross-compound Corliss engines, the basis of com-

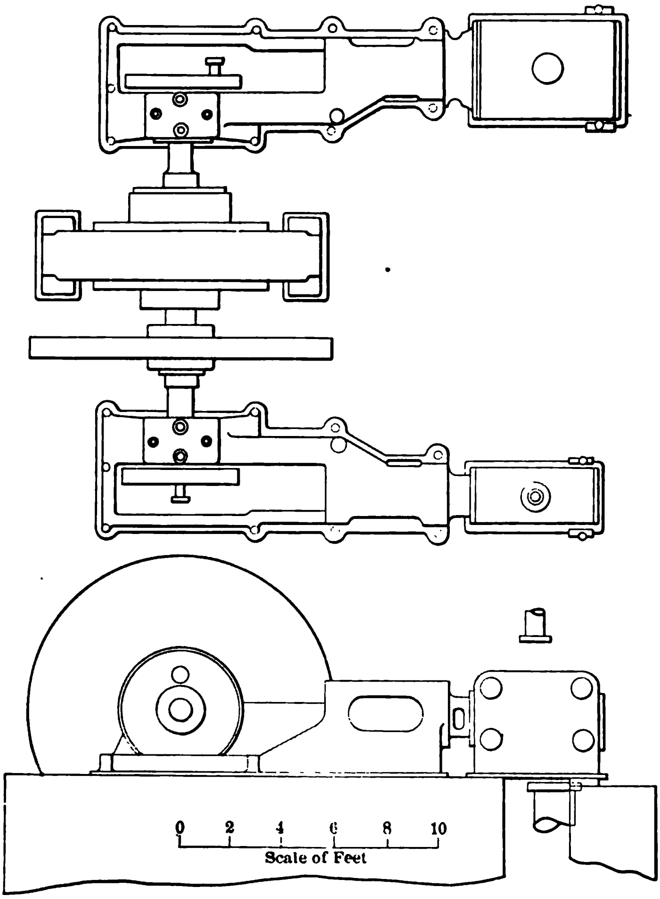


Fig. 5. Plan and Elevation of 500 Kw. Corliss Engine.

parison being a 1,000 Kw. unit, including the direct-connected generator, the engine cylinders being 28 and 56 by 48 inches, 95 revolutions.*

*In pamphlet by Edw. H. Sniffin, issued by The Westinghouse Machine Company.

Fig. 2 is a comparison of the famous Reynolds vertical-horizontal engine of 5,000 Kw., and a Curtis turbine of the same power.

The most favorable case for the steam engine is to be had by selecting high- and medium-speed engines for comparison with

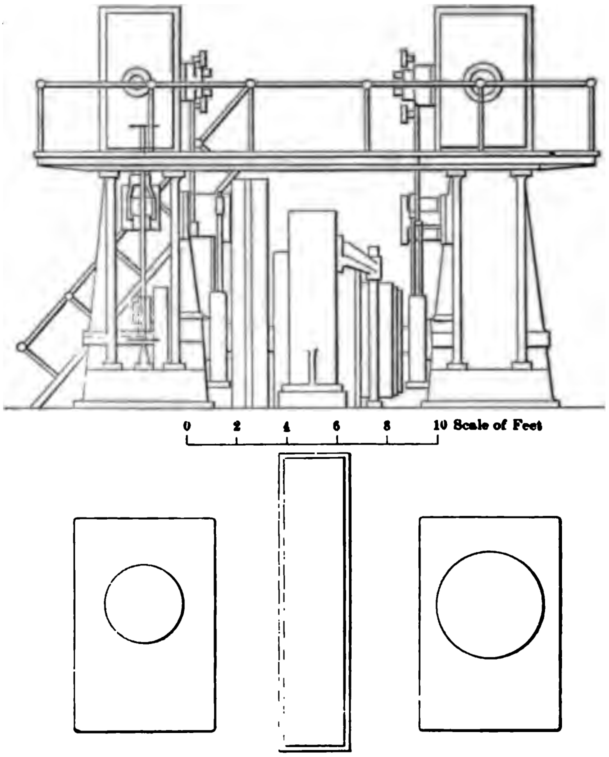


Fig. 6. Plan and Elevation of 500 Kw. Vertical Engine.

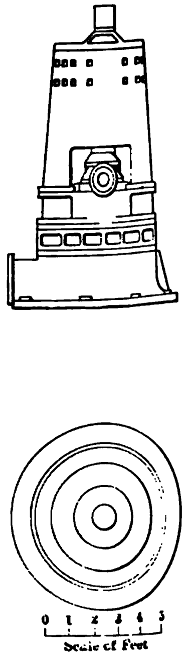


Fig. 7. Curtis Turbine, of 500 Kw.

turbines of corresponding power. In Figs. 4, 5 and 6 are three different styles of engines of 500 Kw. capacity, two of which are horizontal machines and are to be compared with the 500 Kw. Westinghouse-Parsons turbine, Fig. 3, while the third is a vertical machine comparable with the 500 Kw. Curtis turbine, Fig. 7. All

of the diagrams, for both vertical and horizontal machines are drawn to the same scale.

Fig. 4 is a plan and elevation of a McEwen compound high-speed engine designed to run non-condensing and drive a 400 Kw. A. C. generator; but when running condensing it is ample for a 500 Kw. generator. The plan view shows the top of the foundation, which is six inches larger all around than the engine bed.

The vertical engine, Fig. 6, is a cross-compound Shepherd engine, medium speed.

TABLE I.

DIMENSIONS OF HORIZONTAL PARSONS TYPE TURBINES WITH GENERATORS.

Kw	R. P. M.	Length.	Width.	Height.	Weight, lb.
	60 cycles 2	*or 3 phase.*	*Not over*	*6600 volts.*	
300	3,600	23 ft. 4 in	4 ft. 0 in.	5 ft. 1 in.	36,500
500	3,600	24 ft. 6 in	4 ft. 0 in.	5 ft. 1 in.	40,000
750	1,800	27 ft. 0 in.	5 ft. 10 in.	5 ft. 9 in.	65 200
1,000	1,800	29 ft. 0 in.	6 ft. 10 in.	6 ft. 6 in.	81,500
1,500	1,800	31 ft. 10 in.	6 ft. 10 in.	6 ft. 6 in.	103,000
2,000	1,200	34 ft. 9 in	9 ft. 2 in.	8 ft. 2 in.	138,000
3.500	900	35 ft. 9 in.	10 ft. 6 in.	9 ft. 4 in.	237,000
5,500	720	46 ft. 0 in.	11 ft. 4 in.	10 ft. 6 in.	417,000
	25 cycles 2	*or 3 phase.*	*Not over*	*6600 volts.*	
300	1,500	23 ft. 6 in.	6 ft. 4 in.	5 ft. 9 in.	51,000
500	1,500	24 ft. 6 in.	6 ft. 4 in.	5 ft. 9 in.	56.600
750	1,500	28 ft. 3 in.	6 ft. 4 in.	6 ft. 0 in.	75,500
1,000	1,500	29 ft. 9 in.	7 ft. 6 in.	6 ft. 9 in.	104.200
1,500	1,500	32 ft. 6 in.	7 ft. 6 in	7 ft. 0 in.	131,000
2,000	1,500	33 ft. 0 in.	9 ft. 0 in.	8 ft. 0 in.	166,000
3,500	750	42 ft. 5 in.	11 ft. 8 in.	10 ft. 5 in.	356.000
5,500	750	46 ft. 10 in.	13 ft. 2 in.	11 ft. 6 in.	460,000
7,500	750	50 ft. 10 in.	13 ft. 3 in.	11 ft. 6 in.	511,000

Tables of Dimensions.—Table I. gives the approximate floor dimensions and height of horizontal turbines of the Parsons type, with their generators, as made by the Allis-Chalmers Company. In Table II. are the over-all floor dimensions of horizontal, cross-compound Corliss engines, as built by the same firm. These two tables will enable the reader to make an approximate estimate of the saving in space to be effected by installing a turbine of the horizontal type. To facilitate comparison of the two tables, column four was prepared, giving the approximate generator capacity of the different sizes of engines. These values were

worked out by first estimating the indicated horse-power by the rule-of-thumb method of squaring the diameter of the low-pressure cylinder and dividing by two; and then estimating the

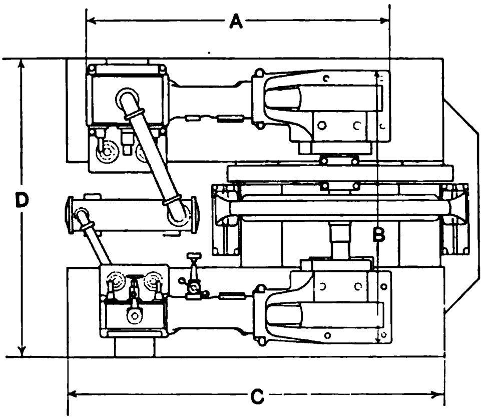


Table II.

Dimensions of Horizontal Cross-Compound Corliss Engines.

Size of Cylinder. Diam. H. P.	Diam. L. P.	Stroke.	Approx. Capacity Generator in Kw.	A	B	C	D
12	24	30	200	18 ft. 3 in.	15 ft. 7 in.	22 ft. 0 in.	19 ft. 0 in.
12	24	36		18 ft. 9 in.	16 ft 8 in.	23 ft. 0 in.	19 ft. 6 in.
14	28	30	300	18 ft. 4 in.	15 ft. 8 in.	24 ft. 6 in.	19 ft. 0 in.
14	28	36		18 ft. 10 in.	16 ft. 8 in.	25 ft. 0 in.	19 ft. 6 in.
16	32	36	400	19 ft. 7 in.	17 ft. 0 in.	25 ft. 6 in.	20 ft. 0 in.
16	32	42		22 ft. 3 in.	17 ft. 6 in.	27 ft. 6 in.	20 ft. 6 in.
18	36	36	500	19 ft. 9 in.	17 ft. 5 in.	27 ft. 0 in.	20 ft. 6 in.
18	36	42		22 ft. 6 in.	16 ft. 0 in.	28 ft. 6 in.	21 ft. 6 in.
18	36	48		25 ft. 3 in.	18 ft. 7 in.	30 ft. 0 in.	22 ft. 3 in.
20	40	36	500	20 ft. 0 in.	19 ft. 6 in.	26 ft. 6 in.	22 ft. 6 in.
20	40	42		22 ft. 6 in	19 ft. 10 in.	29 ft. 0 in.	23 ft. 0 in.
20	40	48		25 ft. 0 in.	20 ft. 5 in.	31 ft. 0 in.	23 ft. 9 in.
22	44	36	750	20 ft. 9 in.	19 ft. 10 in.	27 ft. 6 in.	23 ft. 6 in.
22	44	42		23 ft. 3 in.	21 ft. 2 in.	30 ft. 0 in.	24 ft. 3 in.
22	44	48		25 ft. 8 in.	21 ft. 4 in.	32 ft. 6 in.	25 ft. 0 in.
24	48	42	750	24 ft. 6 in.	22 ft. 3 in.	30 ft. 6 in.	25 ft. 2 in.
24	48	48		27 ft. 0 in.	23 ft. 0 in.	33 ft. 0 in.	25 ft. 0 in.
26	52	42	1,000	24 ft. 8 in.	23 ft. 8 in.	30 ft. 6 in.	23 ft. 10 in.
26	52	48		27 ft. 2 in.	24 ft. 4 in.	33 ft. 0 in.	27 ft. 6 in.
26	52	60		32 ft. 2 in.	24 ft. 10 in.	38 ft. 0 in.	23 ft. 0 in.
28	56	48	1,000	27 ft. 6 in.	25 ft. 10 in.	34 ft. 6 in.	29 ft. 6 in.
28	56	60		32 ft. 6 in.	26 ft. 4 in.	39 ft. 0 in.	30 ft. 0 in.
30	60	48	1,250	27 ft. 9 in.	26 ft. 4 in.	35 ft. 0 in.	30 ft. 6 in.
30	60	60		32 ft. 9 in.	27 ft. 4 in.	40 ft. 0 in.	31 ft. 6 in.
32	64	60	1,500	33 ft. 0 in.	28 ft. 10 in.	41 ft. 0 in.	33 ft. 0 in.
34	68	60	1,750	33 ft. 0 in.	29 ft. 1 in.	43 ft. 0 in.	34 ft. 10 in.
36	72	60	2,000	33 ft. 6 in.	29 ft. 6 in.	44 ft. 0 in.	35 ft. 6 in.

net kilowatt capacity as two thirds of the indicated horse-power. The above rule for indicated horse-power is on the basis of 600 feet piston speed. In the table the same values have in some instances been given to engines of different sizes. In such cases the value ascribed to the smaller engine is on the basis of about 700 feet piston speed.

Space Necessary for Condensing Apparatus. — Comparative

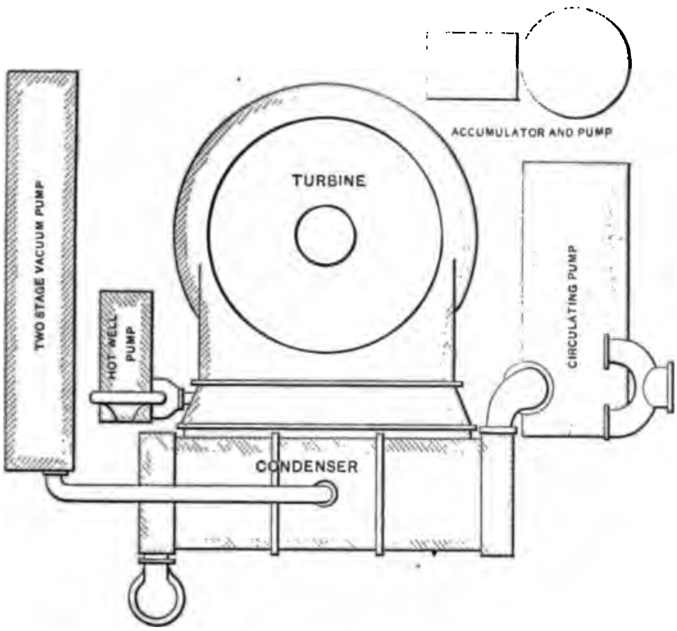


Fig. 8. Arrangement of Condensing Apparatus for Curtis Turbine.

figures of space required for power units are of but little value, unless the room occupied by the condensing apparatus is also taken into consideration. In the engine plant, which is usually equipped with jet or barometric condensers, the percentage of room is very small and can be easily estimated, because of the simplicity of the apparatus.

The condensing apparatus for turbine plants is fully described in Chapter XIX. If a surface condenser is used, it must be at least double the size required for a reciprocating engine and a

corresponding increase in the capacity of the circulating pump and piping. In order to maintain a high vacuum the air pump employed in engine practice is usually replaced by two pieces of apparatus: the hot-well pump, which removes the water of condensation, and the dry-air pump, which exhausts the air and vapor from the condenser. The dry-air pump is frequently made in the form of a two-stage air pump driven by a steam cylinder with Corliss valve gear, making a large and complicated piece of apparatus.

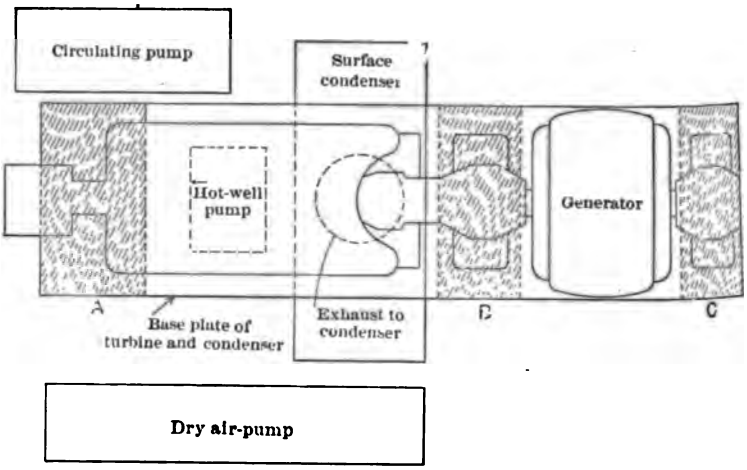


Fig. 9. Condenser Arrangement for Parsons Turbine.

It is difficult to estimate offhand the space required for turbine condensing outfits, because considerations of convenience in pipe connections often make it advisable to widely separate the parts of the equipment. Several diagrams are shown, however, indicating the relative floor space occupied by turbines and their condensing plants, where the condenser and auxiliaries are compactly arranged.

Condensers for Curtis Turbines.—In Fig. 8 is a layout of condenser and auxiliaries for a Curtis turbine, sketched roughly from a blue print furnished by the Alberger Condenser Company. The accumulator and pump in the upper right-hand corner are for the step bearing of the turbine and have nothing to do with

the condenser. Sometimes the condenser is built into the turbine base, in which case space would be required only for the circulating and the hot-well and vacuum pumps. It is not good policy to crowd the condensing apparatus too closely about the turbine, because the machinery will not be accessible for inspection and repairs, and room must always be provided for removing the condenser tubes.

Condensers for Horizontal Turbines.—It is the usual plan to place the condenser and part of the auxiliaries in the basement; and in the case of a horizontal turbine advantage can be taken of the fact that a foundation is not required under its full length, by

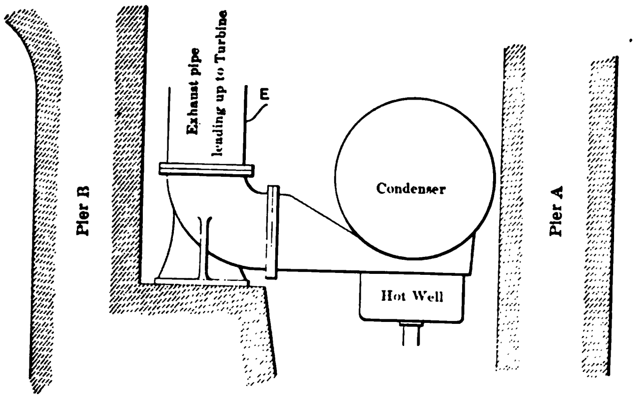


Fig. 10. Connection to Condenser.

utilizing part of the space under the turbine for the condenser. The dry-air pump is so complicated a piece of apparatus that it is better to place this on the turbine room floor, instead of in the basement, to ensure its having the same care and attention, and being kept in as good condition as the turbine itself.

In Fig. 7, Chapter IXX., is one of the most compact arrangements of condenser and pumps that can be devised. The space occupied, including room for removal of condenser tubes, is less than twice the area of the baseplate of the turbine and generator.

A plan that meets with much favor is to place the condenser crosswise of and directly underneath the turbine as in Fig. 9, herewith. The concrete piers supporting the turbine are located

at *A, B* and *C,* as indicated by the dotted sections. An arch would be sprung between *B* and *C* and I-beams running from *A* to *B* would be used to stiffen the baseplate at this point. If the condenser is of the downward-flow type, receiving steam at the top, it would be placed directly under the exhaust nozzle of the turbine, as indicated, and the various pumps could be grouped as shown.

If a counter-current condenser is used, receiving steam at the bottom, it would be necessary to locate it to the left, next to pier *A,* Fig. 9, using an elbow in the exhaust pipe. The location and

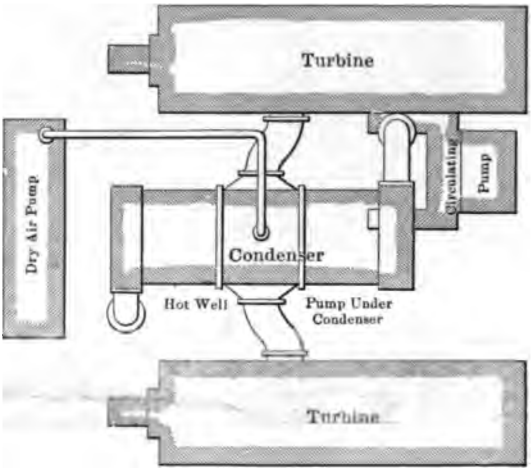


Fig. 11. Two Turbines served by one Condenser.

arrangement are indicated in Fig. 10. A similar arrangement of piping would be adopted if it were desired to set the condenser parallel with the turbine and alongside of the turbine foundation, as is often done. Fig. 8, Chapter XIX., shows a turbine with condenser located underneath.

Frequently space is economized, as well as first cost of apparatus, by having one condenser serve two turbines, placing the condenser between them, as in Fig. 11. This makes one of the most compact arrangements and is satisfactory for small and medium-sized units.

From the foregoing it appears feasible to arrange the con-ensing apparatus to come into an area equal to two to three imes the space occupied by the turbine and generator, in the ase of horizontal turbines; and an area of four to five times the pace required for the foundation of vertical turbines, with due egard for the removal of condenser tubes.

Enlargement of Plant.—The possibilities of the turbine as a means for the enlargement of an existing plant, is well illustrated y the turbine installation of the B. F. Goodrich Company, Akron,). The plan of the engine room is as in the accompanying illus-

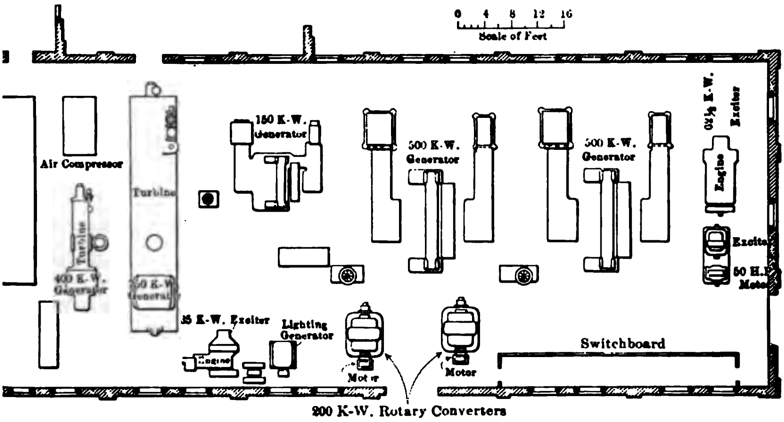


Fig. 12. Engine Room with Turbine Additions.

ration.* There were originally the cross-compound engines and enerators shown, and it was found impossible by any arrange-ment of the machinery to increase the capacity of the plant by ny more than 500 Kw., without extending the building, if the ame type of units was adhered to. Instead, it was decided to nstall two turbines, as indicated in the plan, which doubled the apacity of the plant without disturbing the old arrangement. It vill be evident that another increase in the capacity may be made y replacing the 150 Kw. engine unit by two 400 Kw. turbo

*Originally appeared in *Engineering News.*

units. The power house was originally laid out for 1,650 kilowatts, but without enlarging the building or replacing the large engines it could be made to accommodate 2,950 kilowatts, an increase of 80 per cent. The present engine plant of 1,150 Kw. occupies 2,630 square feet, or 44 per cent of the total. The turbine plant of 1,150 Kw. occupies 980 square feet or 16 per cent of the total. The balance of the space is occupied by other apparatus.

Comparative Cost of Turbine Outfits and Their Maintenance.

The cost of complete turbine and engine outfits is practically the same (exclusive of land and buildings), and such differences as exist will be found to be no greater than often met with in the cost of different engine equipments of the same power. The selling price of turbine outfits is governed by the price of engine outfits rather than by the cost to manufacture.

It will be of interest to compare cost figures for the apparatus of two plants of equal size, one engine-driven and one turbine-driven. Such an itemized statement will show the distribution of expense, illustrating how certain factors entering into the engine costs, such as the foundations and generator, are offset by other items, such as the turbine condensing system. The data will also assist the reader in making his own preliminary estimates.

Example for Comparison.—Assume the case of a 750 Kw. turbine and generator operating with 150 pounds steam pressure and 100 degrees superheat. The generator to be 60-cycle, 3-phase, 2,300 volts.

As the equivalent of this a firm of engine builders have proposed to supply a cross-compound 24 and 50 by 42 engine, 100 revolutions, operating with 150 pounds pressure, saturated steam. The engine is rated at 1,200 horse-power, or about 1½ times the Kw. capacity of the turbine. This is ample to allow for the losses in the engine and generator when comparing the indicated horse-power of an engine with the net output in kilowatts of a turbo-generator. The proportions of the engine are generous, giving ample power for overloads. The generator is to be 60-cycle, 72-pole, 3-phase, 2,300 volts.

Estimate for Turbine Outfit:

Turbine and Generator.—750 Kw., 150 pounds steam pressure, 100 degrees superheat; generator, 60-cycle, 3-phase, 2,300 volts. Price, $22,500.

Exciter.—Engine-driven, $1,500.

Surface Condenser.—Four square feet cooling surface allowed per kilowatt, or 3,000 square feet; 60 to 70 pounds cooling water per pound steam. Condenser equipped with 12-inch centrifugal circulating pump, driven by an 8 by 8 engine, and an air pump, steam driven. Price, $5,200.

Barometric Condenser.—If a condenser of this type is preferred it should be capable of handling 60 times the volume of condensed steam. Price $4,000.

Erecting Condenser.—Two hundred dollars.

Foundation.—Of concrete, 12.5 feet deep. At $6 per cubic yard (a mean value), the cost would be $250.

Superheater.—Estimate on the basis of $2 per boiler horse-power for 50 degrees superheat and $2.60 per boiler horse-power for 100 degrees superheat. One boiler horse-power is the capacity to evaporate 34.5 pounds of water from and at 212 degrees F. The boilers necessary to generate steam for high-grade compound engines or turbines may therefore safely have less than one-half the rated power of engine or turbine. In the present case 500 horse-power boiler capacity will meet the requirements, and the superheater for 100 degrees superheat would cost $1,300.

Estimate for Engine Outfit:

Engine.—Cross-compound, 24 and 50 by 42; 100 revolutions; 150 pounds saturated steam. Price $14,600, or $20 per Kw.

Generator.—Alternating current; 60-cycle; 72-pole; 3-phase; 2,300 volts. Price, $8,950.

Exciter.—Engine-driven. Price, $1,500.

Surface Condenser.—Allow 10 pounds steam condensed per hour per square foot cooling surface, and 35 pounds cooling water per pound steam. Direct-acting pump with circulating and air cylinders. Price, $3,000.

Barometric Condenser.—If preferred, allow 40 to 50 pounds water per pound steam. Centrifugal pump steam driven. Price,

$2,500. An ordinary jet condenser could be installed for $2,000.

Foundation.—Concrete, 12.5 feet deep; 276 cubic yards, $6 per yard. Price, $1,656.

Erecting and Freight on outfit complete, $2,500.

Apparatus in Common.—The feed pumps, switchboard, stack and boilers, superheater excepted, would be the same for either type of plant. Water tube boilers, delivered and erected, cost $14.50 per boiler horse-power. The piping, exclusive of exhaust, would cost practically the same in either case, and may be estimated at $8 per boiler horse-power. The expense of exhaust piping will depend upon the location of condenser, which, in the case of the turbine is often connected directly to the turbine. Smaller water piping is required for the condenser with an engine than with a turbine.

Summary.—Assuming the surface condenser for each type of apparatus and tabulating the *items that differ* in the two types, we have:

Turbine.		*Engine.*	
Turbine and Generator,	$22,500	Engine,	$14,600
Surface Condenser,	5,200	Generator,	8,950
Erecting Condenser,	200	Surface Condenser,	3,000
Foundations,	250	Foundation,	1,656
Superheater,	1,300	Erecting,	2,500
	$29,450		$30,706

Ordinarily the surface type of condenser would not be installed with the engine, thus reducing the cost somewhat.

General Figures.—Taking the above figures, with the omission of the superheater cost, we have, for a 750 Kw. turbo-generator, surface condenser, foundation and installation, $28,150, or $37 per Kw. The engine figures are $40 per Kw. To compare with this, the actual cost of two larger units will be given. One was a 1,500 Kw. turbo-generator, which, with surface condenser, foundation and installation, cost $30.20 per Kw. The other was an 1,100 Kw. cross-compound engine, jet condenser, foundation, and installation, costing $32.40 per Kw.

The cost of turbo-generators per Kw. (60 cycles) is approximately as follows: 3,000 Kw., $20; 1,500 Kw., $24; 750 Kw., $30; 500 Kw., $32.

Surface condensers for high vacuum cost, with necessary auxiliaries, from $7 to $10 per Kw.; and barometric jet condensers from $5 to $6 per Kw. Two additional examples of condenser costs will be cited.

A 1,550 square foot surface condenser for a 750 Kw. engine; 26-inch vacuum (2 square feet per Kw.); direct-acting pump underneath condenser and centrifugal circulating pump and engine at end of condenser. Price, f. o. b. factory, $2,650, or, say $3.50 per Kw.

A 2,000 square foot surface condenser for a 500 Kw. turbine; 28-inch vacuum (4 square feet per Kw.); Edwards air pump; centrifugal circulating pump and engine. Price, f. o. b. factory, $3,852, or, say $7.50 per Kw.

Cost of Land and Buildings.—No estimate of turbine costs is of any value whatever without taking into consideration the cost of land and buildings. In comparing turbine and engine costs the important item to be considered is the investment for real estate in the two cases. Outside of the operating room the space required for the plant will not be materially different, whichever type of apparatus is installed. The turbine room of the turbine plant, however, need be only about one half the size of the same room in an engine plant, and knowing the cost of building and the value of land per square foot in any community, the saving can be very quickly arrived at in the rough. Thus, take the case of a plant with four 750 Kw. units. By the aid of Tables I. and II. and a little figuring with pencil and paper, it will be found that an engine room of 7,600 square feet and a turbine room of half this, or 3,600 square feet will be ample for all the power-generating machinery. At $5 per square foot the land saving would be $18,000 and the saving in the building might easily be as great, depending entirely upon the style adopted.

Cost of Maintenance and Operation.—In a paper before the American Institute of Electrical Engineers, January, 1896, Henry G. Stott, superintendent of motive power, Interborough Rapid Transit Company, New York, gives a careful analysis of power-plant economics. He considers different types of prime movers, including gas engines and combinations of gas engines and turbines and of reciprocating engines and turbines.

In the paper is a tabulation of the relative values of the various items necessary in the maintenance and operation of power plants, and two columns of the table are reproduced herewith. The first column covers a plant with compound condensing reciprocating engines without superheat and is derived from a year's record of actual costs in a plant with 7,500 horse-power Allis vertical-horizontal engines. The turbine compared with this is a 5,000 Kw. unit, which is believed by Mr. Stott to have the

Table III.

Distribution of Maintenance and Operation.
Charges per Kw. Hour.

Maintenance.	Reciprocating Engines.	Steam Turbines
Engine room mechanical	2.57	0.51
Boiler room	4.61	4.30
Coal and ash handling apparatus	0.58	0.54
Electrical apparatus	1.12	1.12
Operation.		
Coal and ash handling labor	2.26	2.11
Removal of ashes	1.06	0.94
Dock rental	0.74	0.74
Boiler room labor	7.15	6.68
Boiler room oil, waste, etc	0.17	0.17
Coal	61.30	57.30
Water	7.14	0.71
Engine room mechanical labor	6.71	1.35
Lubrication	1.77	0.35
Waste, etc	0.30	0.30
Electrical labor	2.52	2 52
Relative cost of maintenance and operation	100.00	79.64
Relative investment in per cent	100.00	82.52

best record for economy up to date. It has a flatter steam rate curve than the engine, shows practically as good economy at normal load with saturated steam and a thermal economy 6.6 per cent better with superheated steam. The various turbine items are derived from actual costs.

Turbine Troubles.

In beginning the construction of steam turbines, it was inevitable there should be difficulties which could not be foreseen and which could be overcome only by observing the machines in

operation after they were erected in the different power plants. With the turbine has come a whole series of new engineering problems, apart from the turbine itself. The greatest of these has been the electric generator, which must not only be right, electrically, but must be free from deformation and the consequent destruction of balance at high speeds. This problem has still to be solved by many of the electric companies not directly interested in turbine construction. In 1906 the author visited an engine works where an experimental turbine had been built and was standing idle on the erecting floor, because it had not been possible to secure a satisfactory generator for it.

Other problems have come from the ability of the turbine to utilize high vacuums, which has required a vast improvement in condensing machinery; and from the employment of highly superheated steam, which has made provision necessary in the turbine and connections to equalize the expansion due to high temperatures.

The best testimony upon the success with which these problems have been worked out is to be had from turbine users themselves. Such testimony has been published in the 1905 report of the standing committee of the National Electric Light Association, for the investigation of steam turbines, and is given in condensed form herewith:

Report of Committee of the N. E. L. Association.—In preparing the 1905 report, 95 companies were written to for information. These companies were operating turbines aggregating over 100,000 kilowatts and special efforts were made to obtain a complete statement of troubles experienced with the machines. Replies were received from 59 companies, and it is interesting to note that all seemed well satisfied with their turbines and very few admitted having any trouble with the turbines themselves, independent of the condensers and other auxiliary apparatus.

Of the replies from users of De Laval turbines, one reported some trouble due to the design of the brush holders of the dynamo, which was easily remedied, while another had trouble with the oiling. This was due to the filling up of the spiral oil-

ways, which stopped the flow of oil, and was overcome by the use of special oil, carefully filtered.

One company, operating Curtis turbines, reported a shutdown caused by a worn bearing on the tachometer connected with the latch of the emergency stop valve; also slight trouble from loose laminations in the armature of the generator. On another Curtis turbine the needle valves and the main nozzle valves were warped by the high degree of superheat employed. On still another turbine of this type there was air leakage in the turbine and also trouble from water mixing with the oil lubricating the step bearing.

Several companies having Parsons turbines reported difficulties. One experienced trouble from the oil solidifying into a jelly in some of the bearings. Another reported the repeated cutting out of the throttle valve seat, which was attributed to the peculiar water used in the locality; also excessive vibration caused by the expansion of the exhaust piping, throwing the turbine out of line. This latter was corrected by installing an expansion joint in the exhaust pipe. The field coils of the generator of two Parsons turbines burned out at less than normal load, indicating some error in design. There was also a breakage of one or more of the brass sleeves of the main bearings of these turbines and special attention was required to keep the lubricating system in good order.

Three companies reported breakages of blades in Parsons turbines. In one case where superheated steam was used, sufficient time was not allowed in starting to warm up the machine and maintain the proper clearance between the blade tips and the casing. In another case, where there was said to have been no rubbing of the blades against the casing, many of the blades of the rotor were broken while running. Whatever the cause of this may have been, such accidents are now guarded against by the use of steel lacing to stiffen the blades. The third company had a few blades broken by some foreign substance carried into the turbine through the steam pipe. This did not affect the operation of the turbine, which continued to run until it was convenient to repair it.

It will be seen from the above that the accidents reported are

ll of a minor character, with the exception of the blade failures. Even when making allowance for the reticence of the firms interrogated, the difficulties experienced must be admitted to be very few and comparatively insignificant.

Danger from Water.—It is claimed for the turbine that it is not injured by water coming over from the boiler in case of excessive priming. Instances are on record where a slug of water has suddenly entered a turbine, bringing the rotating member almost to a standstill, without injury to the machine. Destruction of the blading has also occurred from this cause in some cases, but this is practically impossible except in machines like the Rateau or certain types of Parsons turbines in which the outer ends of the blades are unsupported; and even in these machines the high-pressure blades are so short that damage from water mingled with the incoming steam seldom results. Breakage would be more likely to occur if water should set back from the condenser through some difficulty with the pumps. At the low-pressure end of the turbine the blades are long and slender, and running as they do at very high speed, sudden contact with the water might strip off the last row. Further damage seems to be prevented, however, by the next row of fixed guide vanes, which divide the water into small streams and thus protect the other rotating members. Compared with the breakages that so often occur from water in a steam engine cylinder, turbine troubles from water seem insignificant.

Distortion of Casing.—In Parsons turbines there has been trouble from the distortion of the casing, resulting in the moving and stationary parts coming in contact, tearing out some of the blading. Trouble has been experienced from the casing arching upward under the effect of superheated steam, on account of the top of the cylinder expanding more than the bottom, this being due to the fact that the shell was not made symmetrical, sometimes having ribs or heavier parts at the bottom than at the top. This has been remedied in later machines by more care in the design.

Another cause for distortion has been the "pull" of the condenser at the low-pressure end, due to the vacuum. Inasmuch as a high vacuum is carried, at which pressure steam has a high

specific volume, the opening to the condenser is necessarily of unusually large dimensions and atmospheric pressure distributed over this opening produces a heavy stress. Under the most approved form of construction the exhaust nozzle leading from the turbine passes down through the pedestal at the low-pressure end of the turbine, thus placing this stress directly upon the foundation in so far as possible.

A corrugated copper expansion joint is also placed in the exhaust piping just below the turbine outlet, to compensate for unequal expansion and for any change in the relative positions of condenser and turbine, due to settling of foundations.

Another method that has been tried consists in bolting the condenser flange rigidly to the casing, with the condenser under the casing. The base of the condenser rests on a flexible foundation of springs, sufficient to relieve the turbine of the weight of the condenser, but allowing it to go and come with the turbine. The condenser is thus to all intents and purposes a part of the turbine, supported by the turbine foundation, and it will be evident that the "pull" of the vacuum will have no more tendency to distort the casing than will the pressure at any other part of the casing.

Stripping the Blades.—The most serious accident that can befall a turbine is that mentioned under the last heading, of the rotating and stationary parts coming together and the stripping of the blades. This trouble has been experienced to a greater or less extent in turbines in which the blades have no protection or support at their outer ends. In the early days of the Parsons turbine there was an endless amount of trouble from blade failures. The blades broke, not only when the rotating and stationary parts came in contact, but from no visible cause, one theory being that rapid vibration of the rotating member produced repeated stresses in the blades, leading to their rupture.

Lately a great deal of attention has been given to the blade question. Where the material is an alloy, the composition is selected with due regard to strength and ductibility as well as resistance to erosion. Several manufacturers have adopted steel of high grade. In turbines patterned after the original Parsons type, in which the outer ends of the blades have no support, a

steel lacing is employed, twisted and interwoven between the blades near their outer ends. By these means, and the removal of causes leading to the rubbing of the blades of the rotor against the stationary parts, blade troubles have been almost entirely eliminated, where the machines have proper care in the power plant.

A more substantial blade support is desirable, however, and a marked improvement has been made in the construction covered by the Fulleger and Sankey patents, already described in connection with the Allis-Chalmers and Willans and Robinson turbines. The shroud ring employed in this type effectually shields the blades against destruction, even when there is rubbing contact or a foreign substance enters the turbine.

An interesting example of the effectiveness of this is afforded by a 5,500 Kw. unit installed at the Kent Avenue power house of the Brooklyn Rapid Transit Company, by the Allis-Chalmers Company. When the turbine had been running several days after it was first started, the casing was opened for inspection. It was found that the blading had been rubbing hard against the casing for about one third the length of the turbine, but without in anyway injuring the machine. Later the blading had a still more severe test. By some means a large jackknife had been left inside the turbine. One of the knife blades had gotten between the spindle and the shroud of the first row of stationary blades and, acting like a lathe tool, had cut into the body of the drum for a width of about three eighths and a depth of about three sixteenths of an inch. This cut loosened the calking strip which held the ring of blades in place and the latter, under the influence of centrifugal force, had bent outward, so that the channel-shaped shroud ring had rubbed hard in the bore of the cylinder, and the flanges of the ring had been worn down almost to the heads of the rivets which hold the ring to the blades. Not a single blade, however, had become loosened or injured.

In the Curtis Turbine the shroud ring, and also the base ring of the bucket segments, are wider than the blades themselves, thus effectually protecting them. The construction in this respect is an improvement over the earlier form used.

Blade Erosion.

During the lifetime of a reciprocating engine, there is continually increasing steam leakage, because of the wear of the valves, piston rings and cylinder.* At best this loss is considerable, and in order to maintain the economy of the engine the valves must occasionally be scraped to their seats, the cylinder rebored, and the piston rings refitted.

In the turbine there are no corresponding wearing parts and practically the only deterioration that can affect the steam consumption comes from the cutting action of the steam or water upon the blades. Experience thus far indicates that blade erosion will not prove a serious matter; but the turbine must pass through a longer trying-out period than it yet has to demonstrate whether a drop in steam economy is to be expected from this cause, when a machine has had a long period of service.†

Erosion Caused by High Velocity and Moisture.—To test the tendency of steam flowing at different velocities to erode the surfaces of buckets, Francis Hodgkinson experimented at the Westinghouse Machine Company's plant with hard-drawn delta metal blades exposed to two steam jets. The velocity of one jet was about 2,900 feet per second and of the other about 600 feet per second. The blades were continuously exposed to the jets for 128 hours. Those subjected to the higher velocity were stripped and eroded, while those subjected to the lower velocity were not injured.‡

*In 1904-05 the Steam Research Committee of the Institution of Mechanical Engineers, England, investigated the leakage of valves and pistons of a small slide-valve compound engine. The tests were carried out under all manner of conditions, and the results show that with well-fitted valves the leakage may amount to over 20 per cent and is rarely less than 4 per cent, depending upon the steam pressure, speed, lap of valves, etc. Other types of valves might be either better or worse.

†A 500 Kw. Parsons turbine was installed at the plant of the Cambridge Electrical Supply Company, England. After it had operated about a year, it was tested by Professor Ewing and showed a result of 25 pounds per Kw. hour, normal load; and 24.4 at a slight overload. The factory test of this machine showed a result of 24.1 pounds per Kw. hour. In the later tests, however, besides running with wet steam, the turbine was driving its own air and circulating pumps, and the steam for these was charged to the turbine. In the test at the builder's works the turbine did not drive its own pumps. There have also been some other tests of turbines after comparatively short periods of operation, but as yet no results have been published of tests made after turbines have been in longer operation, say for a period of 10 years.

‡Paper by Francis Hodgkinson before the A. S. M. E. in 1904.

It also appears to be well established that erosion is greatly increased by the presence of moisture in steam, especially when flowing at high velocities, and this may account for the rapid wear in the case of the high-velocity steam in the Hodgkinson experiment. Such action is corroborated by the manner in which steam injectors invariably wear. The steam nozzles of injectors are seldom eroded, although steam flows through them at velocities exceeding 1,500 feet per second; but the combining nozzles, through which the feed water and condensed steam pass at a more moderate rate, are often so badly scored that they must be renewed. Inquiry of several manufacturers of injectors has brought replies showing that but little trouble is experienced with the steam nozzles.

Erosion in De Laval Turbines.—In the De Laval turbine, in which enormous steam velocities are realized, there is occasional cutting of the blades, when the conditions of water or steam are not favorable. This was commented upon in a paper before the A. S. M. E. in 1904, by E. S. Lea, then of the De Laval Steam Turbine Company, who stated that there have been a few instances where buckets have worn out in a year, necessitating replacement. In other cases the wear has been very slight, even in a run of four or five years. The wear affects only the steam inlet side of the buckets and hence does not impair the efficiency to a great extent. In tests upon a turbine of 100 horse-power, where the edge of the buckets had been worn away about one sixteenth inch, the steam consumption was about five per cent higher than with new buckets.

Erosion in Parsons Turbines.—In the Parsons turbine, where steam velocities are low, the trouble from erosion appears to be almost entirely absent, such cutting as occurs being slight and mostly in the low-pressure end, where the steam is moist. Some time ago articles were published in certain technical journals, in which were illustrations of badly scored Parsons buckets, and the impression was conveyed that they were samples of the condition that turbine buckets might be expected to get into. The author succeeded in running down the source of the information and found that the blades illustrated had been taken from a turbine which had become injured by contact of the moving vanes

with the casing, and that the cutting had undoubtedly been done by particles of steel broken off and blown through with the steam.

To further investigate this important subject letters were written to American engine builders and to a number of engineers in England, where the turbine has been used longer than in this

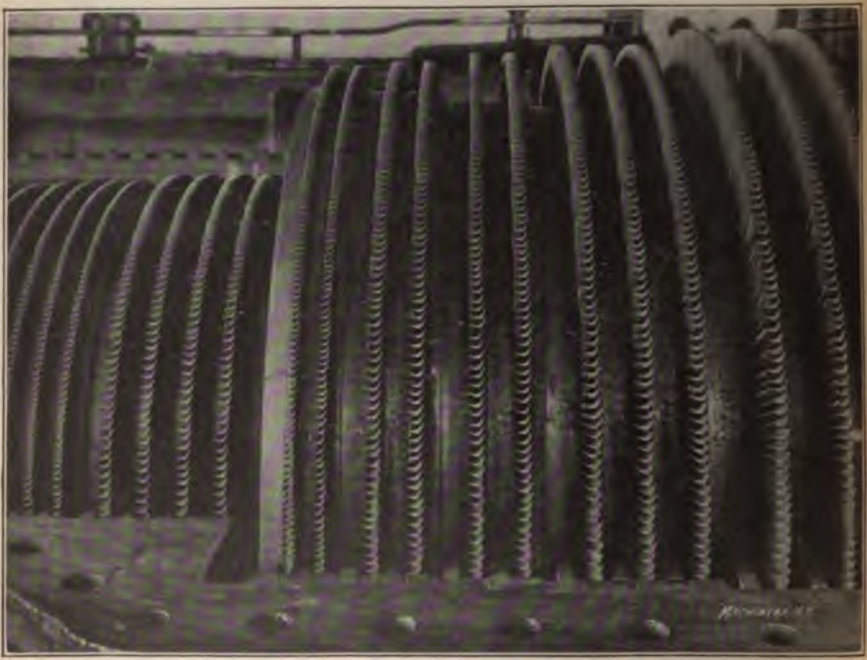


Fig. 13. Appearance of Turbine Rotor after five years' Service.

country, asking for definite information in regard to blade erosion. It was expected that the engine builders, at least, would be well informed upon turbine difficulties; but no information was secured from them, nor from any other source, to indicate that the question of erosion need cause apprehension. Some erosion does occur when the conditions are right for it, even in the Parsons type of turbine, with its low steam velocities;

but engineers in general hold to the opinion that it is not a matter of great moment.

Experience with Westinghouse Turbines.—The first turbine of the Parsons type to be put into practical use in this country was installed at the air brake works, Wilmerding, Pa., in 1899. After it had been in continuous service 24 hours a day for over five years the blades were carefully inspected for wear. Fig. 13 is a view of a portion of the rotor at the low-pressure end, where the steam is the most moist and cutting would be expected, if anywhere. In Fig. 14 is a view of a section of the blading in the upper half of the casing. Both views show the blades to be in good condition, and the company states that the only wear that could be detected was in the case of certain vanes which were slightly out of line with the remaining ones of their particular ring. One of these was broken out and is shown in Fig. 15, photographed beside a new blade for comparison. Such wear as occurred was chiefly at points *a, a.* The edges of the blade were worn to a knife edge. The steam supplied to this turbine is said to have been excessively wet, the feed water extremely acid, and to carry so much sediment that the turbine had frequently to be cleaned out by air blast.

Wear of Curtis Blades.—The only information the author has seen upon erosion in the Curtis turbine has been given by Chas. B. Burleigh of the General Electric Company.* He says:

"The only data we can present are from experience, to the effect that after the most exhaustive tests under most trying conditions no appreciable wear can be detected; and of some 100 turbines of an approximate aggregate capacity of some 103,600 Kw., all of which are in commercial operation and many of which have been in almost continuous commercial operation for nearly two years, we have been unable to detect the slightest evidence of wear.

"I had an opportunity last week of examining the interior of a 2,000 Kw. turbine that had been in service for a year on railroad work, in company with a prominent New England manufacturer, and we were unable to detect the slightest evidence of any change

*Paper on the Curtis turbine before the New England Railroad Club, April, 1905.

Fig. 14. Blading in upper half of Casing.

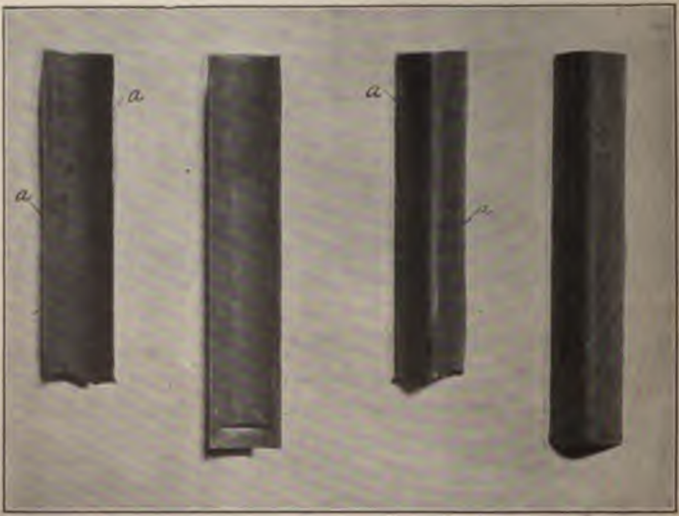


Fig. 15. Old Blades compared with new Blades.

in the appearance of any part of the turbine with which the steam came in contact. . . . . Nor would this be a matter of serious moment if conditions were different, for the reason that if it were necessary to replace every part of a Curtis turbine with which the steam comes in contact the machine is so constructed that this could be accomplished without serious inconvenience and at an expense not exceeding 10 per cent of the first cost."

Composition of Blades.—There is no doubt that the breakage and erosion of blades depends to a considerable extent upon their composition, which, as before stated, has received a great deal of attention. Ordinary bronzes containing tin have their properties too much affected at the temperatures of superheated steam to make them reliable, one of the reasons being that tin melts at 450 degrees F. Brass also weakens at high temperatures, to a less extent, but has been extensively used for blades, the alloy varying from 72 parts copper and 28 parts zinc, to 63 copper and 37 zinc. Its tensile strength, annealed, is about 45,000 pounds per square inch; but by cold drawing this can be increased.

An alloy of about 65,000 pounds tensile strength, which withstands erosion well, consists of 80 parts copper and 20 parts nickel; and it is only slightly affected by temperatures coming within the range of steam temperatures in practice. Steel forgings, which are used more or less for blades, also retain their properties satisfactorily at the temperatures of steam.

One manufacturer uses a nickel bronze, which is said to be a copper-zinc alloy containing a small percentage of nickel and iron, the latter to increase its strength and wearing qualities. This bronze is the result of much experimenting and its makers do not care to give the exact composition.

CHAPTER XVIII

CARE AND MANAGEMENT.

The duties of the engineer of a turbine plant are in most respects like those of the engineer of a plant equipped with reciprocating engines. There are, however, special things to be attended to in order to keep a turbine in good running condition.

First of all it must be remembered that the turbine is a high-speed machine and that if anything is to happen to it it will happen suddenly and almost without warning. A turbine that has frequent inspection and regular care will run day in and day out. But if the oil circulation is allowed to fail, or the step-bearing pump allowed to balk, or other vital part to get out of order through lack of attention, a shutdown is the inevitable result. An engineer must not deceive himself by thinking he can coax a turbine along which he has not kept up in condition. There is no possibility, for example, of nursing a hot bearing on a turbine as so often done with a reciprocating engine.

It is generally held that turbines require less care than reciprocating engines, which is true if by "care" is meant the actual labor expended upon the turbine itself. But if the high-vacuum condensing system be counted in, it is a fair question for argument whether a "rotary engineer" may not be kept just as busy as a "reciprocating engineer."* It needs to be emphasized

*Upon this point C. J. Davidson, chief engineer of power plants, the Milwaukee Electric Railway and Light Company, writes the author as follows: "In our company there is no great difference in the extent and degree or quality of attendance (required by turbines and reciprocating engines), notwithstanding the popular opinion to the contrary. Our experience has been with turbines of the Curtis type. While it may be possible to realize the claims of some of the advocates of turbines relative to their ability to be put quickly in service, it is both our experience and observation that it requires some time after starting for the bucket wheels to find their final running position, due to expansion, and on this account we exercise quite as much care as we would do in warming up a large engine.

"The step-bearing pump must be kept in constant operation or serious results will follow. This means eternal vigilance. Synchronizing two alternators driven by turbines is exceptionally easy; but great precision and consequent care on the part of the attendant is necessary, as a comparatively slight shock will unbalance these machines.

"In general I should say that the modern steam turbine is more refined mechanically and consequently a more delicate piece of apparatus than the reciprocating steam engine, and to insure its reliability of operation probably requires somewhat less labor but correspondingly greater skill than is necessary in case of the engine."

that close watch must be kept of the oiling system, vacuum, step-bearing pump, gland water (the former on the Curtis and the latter on the Parsons), etc., while the turbine is running; and when the turbine is not running, the auxiliaries should be gone over, glands packed, oiling system inspected and cleaned, if necessary, and valves and governor parts inspected for freedom of movement, and for leaks and wear, respectively.

In what follows a few general directions will be given, after which the handling of different turbines will be considered separately.

Starting.—The same care must be exercised in warming up a turbine as a steam engine of corresponding power. Let steam blow through slowly with the turbine standing idle. During this interval start the auxiliaries, first the circulating pump, then the hot-well and dry-air pumps and the oil pump. Finally, start the turbine slowly, later bringing it up to speed. Start the exciter set after the turbine begins to rotate. As in the case of a reciprocating engine, gradual starting will avoid a sudden rush of steam from the boiler, carrying water with it and sweeping along the water of condensation lodged in the piping.

If steam is drawn from a superheater, take particular care in handling the admission valve, for during the warming up the small quantity of steam flowing will become considerably cooled, while the larger volume flowing when the throttle is opened wider will retain its superheat and the turbine will be suddenly exposed to a temperature much higher than that of the warming-up period. In warming up with superheated steam, rotate the spindle slowly for a long enough period to ensure that all parts are brought up uniformly to their normal temperature before bringing up to speed.

The chief concern of the engineer in starting should be to do nothing to produce sudden changes of temperature in the turbine. Where turbines are standing idle part of the time, but are liable to be called into service, it is a good plan to keep them warm all the time by allowing a small quantity of steam to enter continuously, preferably through a by-pass in the throttle. In this way a much quicker start can be made, without danger of a mishap, or of the rotor being out of balance and vibrating. A rapid

start can be made more easily with Curtis than with Parsons turbines and if a Curtis turbine has been kept warm it can be brought up to speed in two or three minutes in an emergency. In general, however, 10 to 15 minutes should be taken in warming up and starting either a Curtis or Parsons turbine, and if the auxiliaries are put into operation at the same time it will usually require about the same interval to get them running regularly.

Shutting Down.—Partly close the throttle before reducing the load on the generator, so the turbine can be brought under instant control in case it should speed up when the load is thrown off. This cannot happen, of course, if the safety stop is operative. After closing the throttle it is well to trip the stop motion to test its action. After shutting off steam close the condenser valve; or if the turbine is connected to an independent condenser, stop the air pump, hot-well pump and circulating pump. It is not uncommon for a turbine rotor, running in vacuum and with no load, to continue to rotate for from 30 to 60 minutes after steam is shut off. The speed can be checked by opening the drains, admitting air to the casing, and by leaving the current on the generator fields. If the turbine has an independent load, instead of running in parallel with others, this can be used to quickly check the speed.

Condensing Apparatus.—The turbine engineer, who has a high-vacuum surface condenser and connected apparatus under his care, will find the turbine itself to be the least source of his troubles. A loss of an inch or two in vacuum in a reciprocating engine plant, where 26 inches is considered a good vacuum, is not a serious matter. But in a turbine plant, where 27 or 28 inches or more are carried, a drop of an inch or two in vacuum means a large increase in steam consumption, as explained in Chapter XIX. It is therefore a much more important matter to keep the condensing system of a turbine plant up to a high state of efficiency than in the case of an engine plant, and it is also a much more difficult matter to do so, because of the greater chance for air leakage through glands, joints and relief valve. It is a temptation to let air leaks go and cover up their existence by pushing the air and circulating pumps, with consequent addi-

tion to operating expenses. But the painstaking engineer will not be satisfied to do business in this way.

More or less trouble is experienced from the carbonization of the oil in the ports and valve chambers of the dry-air pump. One plan for overcoming this is to provide an additional oil cup of the positive-feed type for the air cylinder, and to use this to feed in soap suds along with the oil. The trouble can also be reduced by having the jacket cooling water as cold as possible and forcing a large quantity through the jacket. Mineral oil should be used of the grade designed for gas engines and air compressors.

Changing from Condensing to Non-condensing.—When running condensing, the exhaust end and exhaust passages of the turbine are cool, but if a change is made to non-condensing the temperature of the steam in these passages will rise at once to above 212 degrees and the quantity of steam flowing will also increase. If the change is made suddenly, the turbine will be subjected to wide temperature changes and care must be exercised to shut off the condenser as gradually as possible to avoid this. When changing from non-condensing to condensing, the weight of steam flowing diminishes and the cooling effect will not be as marked as was the heating effect in the other case.

Operating the De Laval Turbine.*

Starting.—Upon first starting, after erecting or after a long shutdown, the bearings should be flooded with oil, the amount being gradually reduced to the normal quantity. The oil reservoirs on the self-oiling bearings should be filled until the oil stands between the red marks on the gauge glass. The small oil valves on the governor valve should be filled with cylinder oil, the valve stems then pressed down, thus allowing the oil to pass into the governor valve. Steam is then turned on, and the governor valve and wheel case allowed to become thoroughly heated. Before doing this, however, the nozzle valves should be opened about a half turn; otherwise they will stick when the wheel case becomes hot. The turbine should be started gradually so as to give the bearings time to heat thoroughly. More time

*Abridged from directions furnished by the De Laval Steam Turbine Company.

is required for this in the larger turbines than in the smaller. As soon as the turbine starts, the self-oiling bearings must be examined to see if the oil rings run properly. If the turbine is running condensing, the condenser should be started first. If starting with no load it is well to start with a low vacuum, say from 24 to 25 inches. As soon as the load is put on, the vacuum should be raised to its maximum.

Shutting Down.—When a machine running non-condensing is to be stopped, the throttle valve should be closed and the lubricator shut off as soon as the machine has come to a standstill. If the turbine is running condensing, and if operating the water and air pumps, either directly or indirectly, the air cock on the exhaust end of the turbine wheel case should be opened before the throttle valve is shut off.

General Care of Turbine.—The usual precautions, with which engineers are familiar, should be taken to keep the oiling arrangements in working order. The sight-feed lubricator must be kept clean and the oil in the self-oiling bearings, and accumulating in the gear case, drawn off and filtered as often as necessary. Particular attention should be given to the oiling of the governor mechanism, and especially the contact surfaces between the governor pin and the plunger on the bellcrank. The high-speed bearings should be removed and examined at intervals. Should a bearing run hot, it should be taken out, the oil-grooves cleaned and if any bright or black spots appear, they should be removed with a scraper.

The strainer above the governor valve, to prevent foreign particles from entering the turbine, should be removed and examined at least once a month.

If the turbine speed is too high, the brass nut holding the governor springs should be screwed out, or if the speed is too low, the nut should be tightened. It is well, every time a turbine is started, to press down the bellcrank, to ascertain that these parts do not stick; and when fully depressed, the governor valve should shut off steam entirely, or at least within a few pounds.

To keep the gears in proper condition, the teeth should be cleaned occasionally when the machine is not running. Kero-

sene and a metal brush are the best for this purpose. The gear case should also be cleaned at the same time and the gears well lubricated.

If, for any reason, the gears have to be taken out of the case, the engineer should secure special directions from the manufacturers, relating to the adjustment of the gears as well as to their removal. The gears need to be kept in perfect adjustment, as their life depends to a considerable extent upon this being done.

Operating the Parsons Turbine.

Starting and Stopping.—It is the rule of many engineers never to allow a condensing engine to run without a vacuum, either in starting or shutting down, if it can be avoided. They follow the same rule in operating turbines as in the case of engines and start the condenser pumps while the turbine is warming up and then bring the turbine up to speed with the vacuum on. If the turbine is a Westinghouse an objection to this method lies in the fact that the glands of this turbine, which are water-packed, do not become sealed until the machine is in operation; and if the turbine is started when subject to vacuum there will be leakage of cold air in through the glands, tending to set up unequal temperature conditions in the turbine.

The author has not learned of any trouble from this cause, but if it is desired to avoid this condition the order of starting is to start the circulating pump while the turbine is warming up; then gradually start the turbine and exciter set. When the speed increases to the point where the load begins to come on, turn on the gland water, start the hot-well pump and dry-air pump, and bring the turbine and exciter up to speed together. In shutting down turn off the water from the glands as soon as the vacuum drops, and open the drip pipes.

At the plants of the Hartford Electric Light Company, Hartford, Conn., where they have had the longest experience with Westinghouse turbines of any company in the country, it is the practice to start up non-condensing until the glands have been closed, or nearly so, before throwing in condensing.

Directions by an Engineer.—W. H. Damon, Springfield, Mass.,

engineer of the United Lighting Company of that city, has written the author regarding the care of Westinghouse-Parsons turbines as follows:

We have three 1,000 Kw. Westinghouse-Parsons turbines connected with jet condenser. They have given us no trouble at all, sometimes running from Sunday to Sunday without stopping.

In starting up we warm up the turbine, start the dry-air pump, and then the injection pump. The turbine is then started slowly, followed by the exciter, which is steam-driven. The turbine and exciter are brought up to speed at about the same time, taking from 10 to 15 minutes to get up to speed. If you start too fast there is too great a vibration.

After the turbine is up to speed and the load on, all the attention it needs is to watch the oil supply on the bearings and the gland water.

The auxiliaries need more care than the turbine. We have had some trouble from the oil carbonizing in the valve chambers of the dry-air pump, stopping up the ports and causing the valves to stick, but have overcome this to a great extent. Otherwise the care of the outfit is simply keeping things clean, cleaning the oil strainers on the turbine, keeping the governor from getting gummed and the pilot valve in good condition. The oiling system should be given close attention, being careful not to pump any air into the system and having plenty of oil in the suction tank.

In stopping close the throttle and shut down the exciter and condensers, being sure to shut off the gland water as the water might otherwise get into the oil as the vacuum falls.

We use an auxiliary oil pump to ensure a good supply of oil on the bearings when the turbine is running at a slow speed.

Care of the Turbine.—In the Parsons turbine the spindle bearings support the weight of the drum and this weight, in connection with the high speed of the journals, causes the bearings to run so hot that the hand can scarcely be held on them. Their high temperature makes necessary the cooling coil for the oil and this must be cleaned as often as required to keep the coil surfaces effective. This alternate heating and cooling of the oil makes some oils, which otherwise would be good lubricants, poorly adapted for turbine work. The heating tends to decompose them and if they contain paraffine this will be deposited on the surfaces of the coil and in the oil passages during the cooling process. Some oils, also, have a tendency to form an emulsion when they become mixed with water, which

might happen in case of leakage from the turbine glands. This emulsion is like jelly and chokes up the coil.

In the general care of the turbine, the governor parts and connections with the primary and secondary admission valves must be regularly inspected to see that they do not become gummed, and it must be seen that the pilot valves work freely and are in good condition. Occasional inspection of the blading is advisable, at which times the blade channels may be cleaned, if required. When the machine is running, give the oiling system close attention, making sure there is enough oil in the suction tank to avoid air being drawn into the system. Try the pet cock on each bearing frequently to see that the oil is circulating properly. In the base of the turbine is an oil strainer which should be removed and cleaned every few days and which can be done while the turbine is in operation. The oiling system and the water supply to the glands and cooling coil are the three things that require regular attention when the turbine is running.

Operating the Curtis Turbine.

Practically the only feature of a Curtis Turbine (aside from the condensing apparatus) which requires care or attention different from and in addition to that which would be given a steam engine is the high-pressure hydraulic system for the step bearing. Double-acting duplex pumps are used, usually in connection with an accumulator, and so much depends upon the maintenance of pressure in the step bearing, and the pump is working under such high pressure, that unusual care must be taken to keep the pumps packed and in good order and to see that they are regularly inspected when in operation.

Directions for Care of Turbine.—The author has talked and corresponded with many turbine engineers in regard to the management of their plants, and among the letters received is one from A. A. Leavitt, engineer of the Gloucester, Mass., Electric Company, in which are the following concise directions for handling Curtis turbines:

1. The air and circulating pumps and all piping connections to the same must receive frequent attention, as it is of the greatest importance that the vacuum carried be as high as possible.

2. The pressure pumps must be kept in first-class condition and piping examined frequently to ensure its being in good condition. As the pressure carried is high, usually about 500 pounds per square inch between pumps and accumulator, and about 200 pounds at the step bearing, and as this step bearing is what carries the whole machine, the importance of attention to this feature is evident. It requires great care in packing the step-bearing pumps to ensure a steady, uniform full stroke.

3. The oiling system must be kept tight, as a small leak will not only make a machine look unsightly, but will materially affect the operation of it by the oil dropping down onto the collector rings and causing sparking.

4. The brushes and collector rings must be kept absolutely clean and perfectly adjusted, to ensure steady voltage; for after they start to spark the voltage will be very unsteady.

5. The governor must be kept in the best possible condition to ensure steady speed. As the governors of these machines all run at high speed the parts wear quite rapidly and this wear should be detected and remedied by making the necessary adjustments.

6. In starting, the turbine should be given time to warm up and the parts expand to working conditions, especially if a high degree of superheat is used. The step-bearing pumps are first started, to give the accumulator time to rise to its position, and then the circulating pump is started, then the vacuum pump and lastly the oil pumps. After these are all working properly the turbine is started. The exciter set is started and the current put on the turbine fields when the turbine is up to about half speed.

7. In shutting down, shut steam off the turbine first, then stop the air pump, the circulating pump, and last the oil pumps and step-bearing pumps. When the turbine has come to rest, it should be carefully gone over and scrupulously cleaned, the same as any dynamo, as there is nothing which will collect dirt any faster than electric machinery and there is no machinery to which dirt is any greater detriment.

Practice at a Large Turbine Station.—At the L-Street station of the Edison Lighting Company, South Boston, Mass., are four 5,000 Kw. Curtis turbines, each with condenser in its base and auxiliary apparatus ranged about the turbine on the same floor level as the turbine. In Fig. 5, page 379, is a view of one of these units, with its group of auxiliaries. Cooling water is supplied to the condenser by a steam-driven centrifugal pump. The wet or hot-well pump is an electrically driven centrifugal pump placed in a pit below the floor level. The air pump has a single cylinder, steam driven. There is a boiler feed pump for each turbine unit, which discharges into a heater where the feed

water is heated by steam from the auxiliaries and from an intermediate stage of the turbine, from which it is taken direct.

The author is enabled to publish a few notes on the practice followed in the operation of the several units at this station.

Starting and Stopping.—The order followed in starting is first to prime and start the circulating pump so there shall be no possibility of overheating the condenser by admitting steam before the cooling water is flowing through. The step bearing and oil pumps are then started; then the wet and dry vacuum pumps; and finally the turbine and exciter and the feed pump. In stopping, the load is taken off when the speed of the turbine has been reduced to the point where the voltage and cycles drop. It is general practice to shut down with the automatic valve instead of with the throttle, to test its reliability. The pumps are shut off in the opposite order from which they are started, but the step bearing pump must of necessity be kept in operation until the turbine ceases to rotate, which requires about ¾ hour at this station.

Condenser Pumps.—In order to prime the circulating pump a small air pump or air ejector must be used, the latter type being employed at this station.

The wet pump is always started before the dry air pump, to remove any water that may have collected in the condenser during the shut-down period, such as might occur if there were a leaky throttle or if the step-bearing water discharged into the condenser, as is the case in this station. When the wet pump has reduced the water to a safe level the dry pump is started. By following this order the danger of injury to the dry air pump, by drawing a large volume of water into the air cylinder, is avoided.

The wet pumps are of the centrifugal type and in their operation it is found important to keep the glands of the impeller shaft well packed to prevent air leaks. Such leaks, if considerable in amount, reduce the effectiveness of the pumps and cause water to accumulate in the condenser. Care is also taken that these pumps do not run without a partial supply of water to act as a lubricant for the rubbing parts.

Oiling System.—The gravity oiling system is employed for all

the main bearings, except the step bearing. Oil is pumped from a receiving tank to an elevated supply tank, from which it flows to a small distributing reservoir at the top of the turbine. The system requires no special attention, therefore, different from that required in engine work.

The lubricant used for the step bearing is water. The step-bearing pumps operate against a pressure of 1,200 pounds, which is reduced to 800 pounds at the step bearing by passing through a pressure reducer of the baffle type. An accumulator is used with each turbine to maintain a steady pressure and to act as a pressure storage in case the pump fails, the capacity being sufficient to hold the pressure for 10 minutes. If a quick start is likely to be required, the accumulator is shut off from the system when the turbine is stopped, so that the step-bearing pressure will be ready at a moment's notice.

No trouble has been experienced with the step bearings in this station. If the water were gritty, however, the bearings would wear down and would need occasional adjustment to bring the turbine rotor into proper position, as determined by clearance indicators on each stage. Much of the foreign matter in the water is removed by the baffle above mentioned, which should be occasionally cleaned. It is found that even with the large accumulators used at this station it is possible to pack them so tight that the plunger will not drop, and it is the practice to test each accumulator daily for freedom of movement by causing the ram to move through its whole range of travel.

Warming up and Synchronizing.—Each of these machines is warmed up by a special by-pass and three admission valves which are electrically and separately controlled and furnish the necessary amount of steam for warming up and also starting the turbine and bringing it up to speed, taking, in this case, only from two to five minutes. The exciter set is started and the field given excitation during the early period of raising the speed. In synchronizing great care is taken to have the machine come into phase while its speed is accelerating instead of falling off. It is common experience that turbines which were once in good alignment and balance may be thrown out of balance by lack of care on the part of the operator in synchronizing. But by following

the method advocated above, severe shocks will be avoided and the balance and adjustment of the parts preserved.

Notes of Experience.—C. E. Stanton, chief engineer of the Union Electric Company, Dubuque, Ia., gives the results of his operative experience with Curtis four-stage 500 Kw. turbines.* The chief difficulties have been in connection with the water supply for the step bearings, the gravity oil supply for lubrication, and the occasional sticking of the nozzle valves. The difficulty in lubrication arose through an air lock formed in the gravity oil tank, allowing air to come into the oil feeder pipe line and interfere with the flow of oil, and was remedied by venting the top of the tank.

All water for the step bearings in his plant passes through a strainer after leaving the pumps, to remove particles that might clog up the passages of the step bearings or injure the latter. The pumps for the service have fibrous packing and if this is left until it loses its elasticity and becomes soft, small particles find their way into the strainer and soon choke the supply of water to the step bearings. Dirt or particles of packing, when once in the system, may find their way into the strainers, even after many days or weeks, and the strainers must therefore be cleaned at intervals of twenty-four hours.

The hydraulic accumulator for the step-bearing system, if allowed to remain in one position for a considerable period of time, was found to rust fast and not drop, even if all the pressure was removed from the system, thus defeating the object for which the accumulator is intended. It therefore must be tested frequently by allowing the ram to drop slowly and then return to its former position, a test that was made each day. One other precaution that should be taken with accumulators for this work is to have some kind of signal, usually a steam whistle, which will blow if the accumulator starts to come down, thus notifying the engineer of the failure of the oil supply.

In the 500 Kw. turbines there are eight main nozzle valves, each with its individual pilot valve, which is electrically controlled. On any load within the rated capacity of the turbine, running condensing, five valves are all that open, leaving three

*Paper presented at meeting of Iowa Electrical Association, April, 1906.

valves which might not open for days at a time. If these are left long, they will corrode and stick and if a heavy overload should come might not open at all—or if they did open they might remain in this position. To obviate these troubles all valves are opened and closed several times each day when starting the turbines. Some difficulty was experienced in securing suitable packing for the main nozzle valves which would stand a high degree of superheat. Metallic packing was not successful and asbestos ring packing is now employed, which is satisfactory, except that the valve stems must be repacked more frequently than would be the case if metallic packing could be used.

CHAPTER XIX

CONDENSING APPARATUS FOR HIGH VACUUM.

One of the advantages of the turbine, from a thermodynamic standpoint, is its ability to utilize a high vacuum and expand steam down to the lowest pressure that can be attained in a condenser. The only restriction to the number of times that steam may be expanded in a turbine is that the passages must be large enough to accommodate the increased volume of the steam at the low pressures. In order to maintain a high vacuum, however, a large, expensive and somewhat elaborate condensing system must be adopted, which requires constant attention to keep in a high state of efficiency. On this account the high-vacuum feature of turbine plants has not worked altogether to their advantage, and condensing outfits have generally given more trouble and have been a greater source of expense than the turbine itself.

Effect of High Vacuum with Steam Engines.—While it is, on the whole, thought desirable to provide a condensing system for steam turbines for a vacuum of about 28 inches, the gain in a steam engine from increasing the vacuum above 26 inches is so slight as not usually to warrant the extra expense. A compound engine, with the usual cylinder ratio of 4 to 1, will expand the steam from 10 to 15 times. The volume of one pound of steam at 150 pounds pressure, absolute, is 3 cubic feet. In expanding 15 times, or to a volume of 45 cubic feet, the terminal pressure in the low-pressure cylinder would be between 8 and 9 pounds, assuming cylinder condensation to be balanced by reëvaporation. When the exhaust valve opened, therefore, the pressure would drop suddenly to that of the condenser, and the only effect of a vacuum higher than that represented by the 8 or 9 pounds pressure in the cylinder would be to reduce the back pressure against the piston during the return stroke.

If the attempt were made to carry expansion too far in a steam engine, the low-pressure cylinder, valves and passages would have to be abnormally large and would offer a great deal of frictional resistance. Under such conditions a point would be reached where

the pressure of the steam would not be sufficient to overcome the frictional resistances, to say nothing of doing useful work, and the expansion of the steam beyond this point would therefore be a dead loss. The increased condensation in the low-pressure cylinder would also be a serious factor.

Below are given the volume of one pound of steam corresponding to different "vacuum" pressures, indicating how impossible it is to utilize these low pressures in the steam engine. To expand

Absolute Pressure.	Vacuum, Inches.	Specific Volume.
½	29	636
1	28	335
2	26	174
3	24	118
4	22	90

steam from 150 pounds to 1 pound absolute, or to 28 inches vacuum, would mean that the volume must increase 111 times. To carry the expansion to this point in a compound engine, the ratio of the cylinders would have to be about 33 to 1; that is, the diameter of the low-pressure cylinder would be 10½ times that of the high-pressure cylinder—quite an impracticable figure.

In the case of the turbine, however, the steam may easily be expanded from 100 to 150 times without encountering any constructive difficulties.

Why a Turbine Derives more Benefit from High Vacuum than an Engine.—Fig. 1 illustrates the expansion of one pound of steam from an initial pressure of 100 pounds to the pressures indicated, and illustrates the difference between the way in which an engine and a turbine benefit from a high vacuum. It shows the work done both before and during expansion, as in an indicator diagram. The section of the diagram marked *a-b-c-d-e* represents that part of the energy of the steam that might be converted into work by a condensing engine operating against a back pressure of four pounds, or a vacuum of about 22 inches. At point *c* expansion has been carried as far as the size of the engine cylinder permits and hence, when the exhaust valve opens, the pressure drops from point *c* to point *d*.

Now assume the back pressure to be reduced to two pounds, and it is evident that the gain in power for the engine would be due simply to the reduction in back pressure represented by the

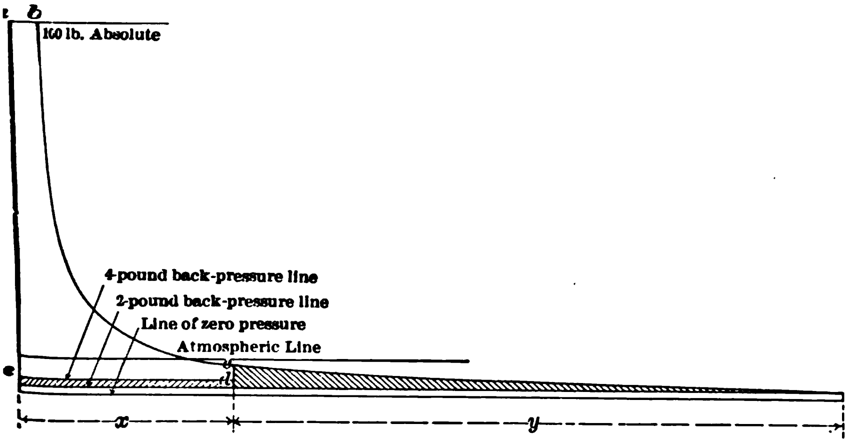


Fig. 1. Diagram Showing How the Turbine Takes Advantage of High Vacuum.

shaded portion having the length x on the diagram. This, it will be noticed, is but a small percentage of the total area of the diagram. In the turbine, however, it is different, since expansion can be caried to the lower back pressure line within the turbine, itself. The turbine is able to utilize the toe of the diagram, indicated by the shaded portion y, in addition to the shaded portion x, while the engine is unable to turn to any account the energy represented by the toe of the diagram.

Theoretical Gain from High Vacuum.—An idea of the theoretical gain can be obtained by referring to a few calculations. Konrad Anderson* compares power values for steam expanding from 60 and 200 pounds, respectively, and finds that the theoretical gain in running condensing, with 25 inches vacuum, over running non-condensing is nearly 100 per cent with steam at 60 pounds pressure, and 50 per cent with steam at 200 pounds pressure. If the vacuum be then increased to 28 inches, the gain with the 60-pound steam will be about 22 per cent and with the 200-pound steam about 18 per cent. This shows that the per-

*Transactions Institute of Engineers and Shipbuilders of Scotland, 1902.

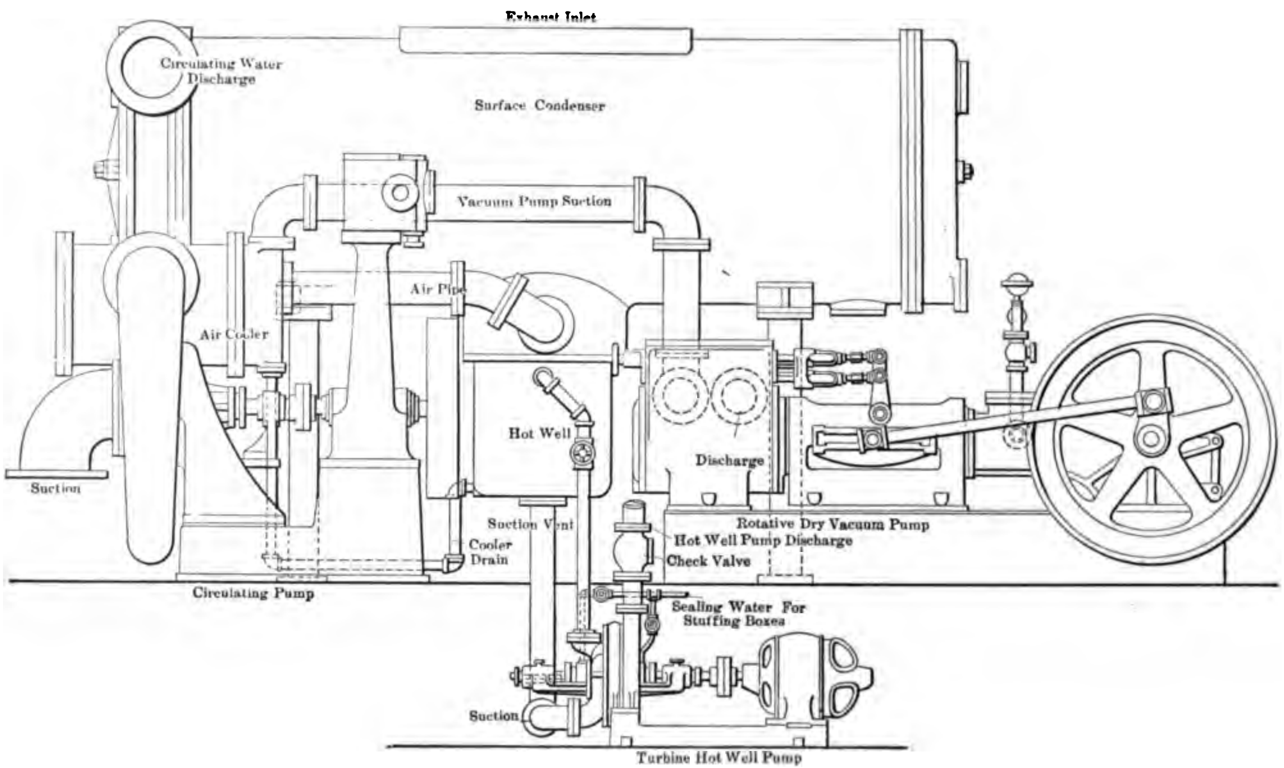


Fig 2. Typical Arrangement of Worthington Condenser and Auxiliaries.

centage gain from running condensing is more with low-pressure than with high-pressure steam and that the gain coming from the last few inches of vacuum is relatively much more than from the first few inches.

This latter fact has been brought out in a striking manner by Ernest N. Janson.* He shows that with the initial and terminal pressures in the same ratio, the kinetic energy of the steam in flowing from a higher to a lower pressure is nearly the same, without regard to what the initial pressure is. For example, supposing the initial pressure to be 105 pounds and steam to expand to one-third this pressure, or to 35 pounds, he finds the kinetic energy developed by the steam to be only 10 per cent more than when expanding from 3 pounds to 1 pound. The figures are as follows:—

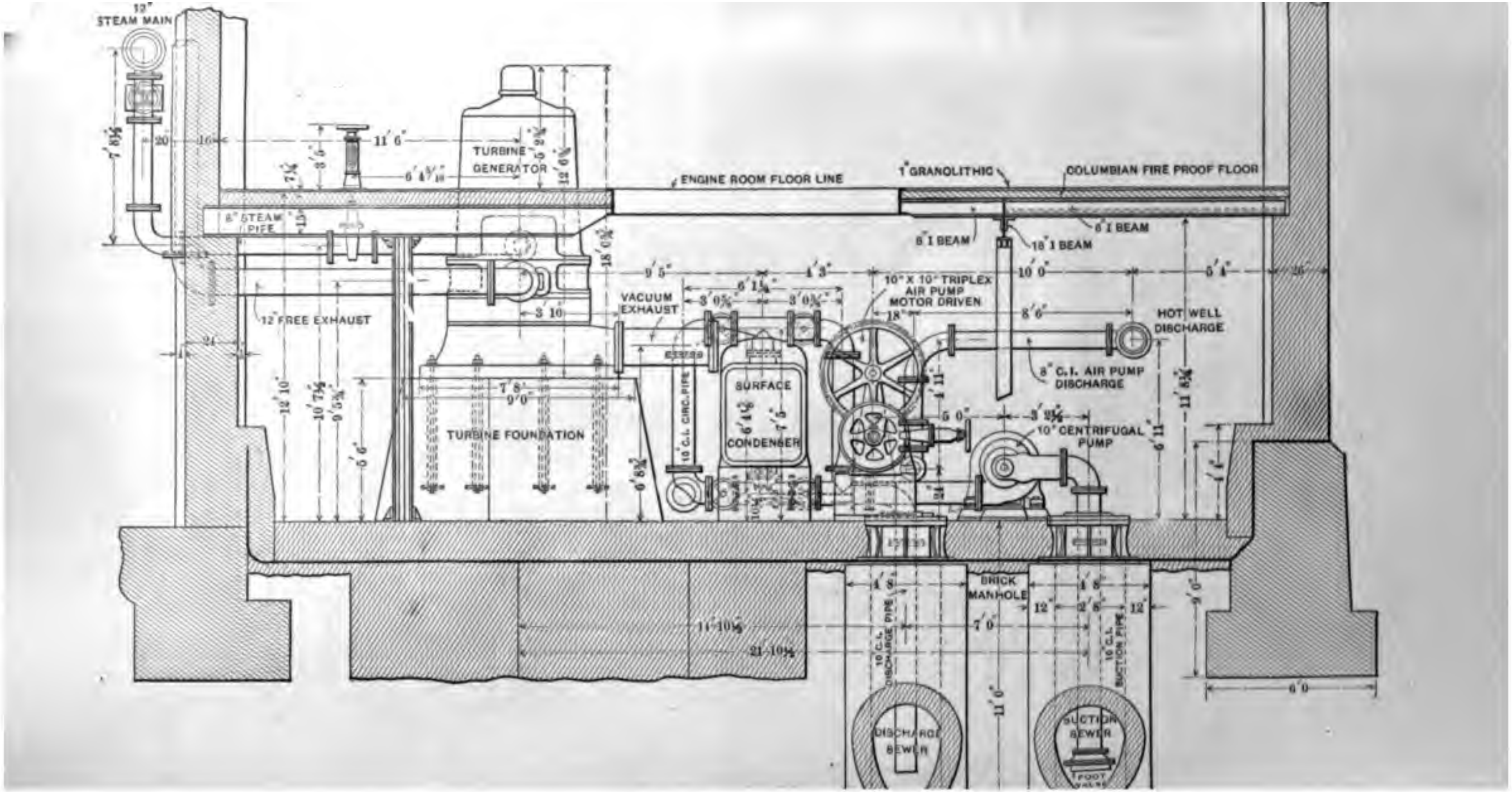
$\frac{p_1}{p_2} = \frac{105}{35} = 3$; velocity = 2,050 ft. per sec.; H. P. per lb. of steam per hour = 0.033.

$\frac{p_1}{p_2} = \frac{3}{1} = 3$; velocity = 1,850 ft. per sec.; H. P. per lb. of steam per hour = 0.027.

Surface Condenser Plants.

The essential features of a condenser plant for high vacuums, as built by the Henry R. Worthington Company, are shown in the diagram Fig. 2. Steam enters at the top of the condenser and is distributed over the tube surface by baffle-plates, while the condensed steam drops down into the hot well at the bottom, where it is discharged by a rotary pump. This pump requires neither valves nor floats and is not subject to vapor binding as are reciprocating pumps. The capacity of the pump is such that it runs ahead of the supply so that the suction pipe is never full; but the discharge pipe is always full and the water presses back against the pump. As long as the latter is in motion, however,

*Article upon steam turbines in the *Journal of the American Society of Naval Engineers.*



12" STEAM MAIN
TURBINE GENERATOR
ENGINE ROOM FLOOR LINE
1" GRANOLITHIC
COLUMBIAN FIRE PROOF FLOOR
6" STEAM PIPE
8" I BEAM
18" I BEAM
12" FREE EXHAUST
VACUUM EXHAUST
10" X 10" TRIPLEX AIR PUMP MOTOR DRIVEN
HOT WELL DISCHARGE
8" C.I. AIR PUMP DISCHARGE
SURFACE CONDENSER
10" C.I. CIRC. PIPE
10" CENTRIFUGAL PUMP
TURBINE FOUNDATION
BRICK MANHOLE
10" C.I. DISCHARGE PIPE
10" C.I. SUCTION PIPE
DISCHARGE SEWER
SUCTION SEWER
FOOT VALVE

the water cannot pass back into the condenser. The condenser is fitted with an air cooler which is frequently applied when high vacuums are to be maintained. This is simply a small chamber containing tubes like a surface condenser. The vapor and air from the condenser pass through this air cooler, where they are cooled by circulating water and their temperature and specific volume thereby reduced. A rotative dry vacuum pump exhausts the air and vapor from the air cooler and maintains a high vacuum. The rotary circulating pump driven by an engine is used for the cooling water. It is usual in installations of this kind to maintain practically a constant supply of cooling water, sufficient to meet the conditions under full load.

Wheeler Condenser and Edwards Air Pump.—In Fig. 3 is an elevation showing one of the 500 Kw. Curtis turbines at the Newport, R. I., station of the Massachusetts Electric Company. This turbine is equipped with a Wheeler condenser and an Edwards air pump made by the Wheeler Condenser and Engineering Company, New York. The construction of this pump is such as to make one of the simplest possible arangements of the condensing apparatus. Fig. 4 is a section of the pump cylinder. It has no foot valves, which require a pressure in the condenser somewhat above that in the pump in order to lift them. The condensed steam flows continuously by gravity from the condenser into the base of the pump and is there dealt with mechanically by the conical bucket working in connection with a base of similar shape. Upon the descent of the bucket the water is projected at a high velocity through the ports into the working barrel; the plunger then rises, closing the ports, and sweeps the air and water before it, causing them to escape through the valve at the top of the barrel. The elimination of the foot valves in this pump enables a higher vacuum to be obtained than with the old style pump, so that 27 or 28 inches can be maintained without the use of an auxiliary air pump.

As indicated in Fig. 3 the condenser is located near the base of the turbine and in front of it are the Edwards air pump and a centrifugal circulating pump, both driven by electric motor. In this plant, as in others arranged according to modern ideas, the suction sewer and discharge sewer for the circulating water are

nearly on the same level. The system of piping leading from the suction sewer, through the circulating pump and condenser, and back to the discharge sewer, thus constitutes a closed circuit, one column of water balancing the other. The sole work of the cir-

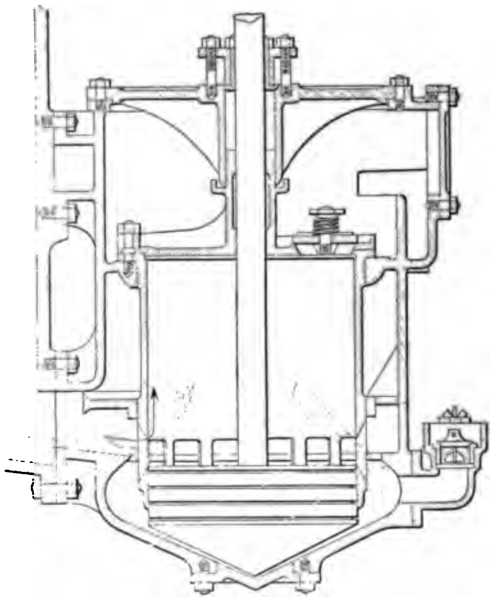


Fig. 4. Cross Section of Edwards Air Pump.

culating pump, therefore, is to overcome the frictional resistance of the water flowing through the piping and condenser tubes.

Curtis Turbine with Condenser in Base.—In Fig. 5 is one of the 5,000 Kw. Curtis turbine units, with its condenser and other auxiliaries, installed at the L-Street station of the Boston (Mass.) Edison Company. In this case the condenser is built into the base of the turbine and forms a part of the unit, while the auxiliaries are on the same floor level as the turbine itself, where they are more accessible. This illustration gives an excellent idea of the quantity of apparatus required to keep the plant in operation, since the feed pumps, heater, hot-well, and accumulator for supplying the hydraulic pressure

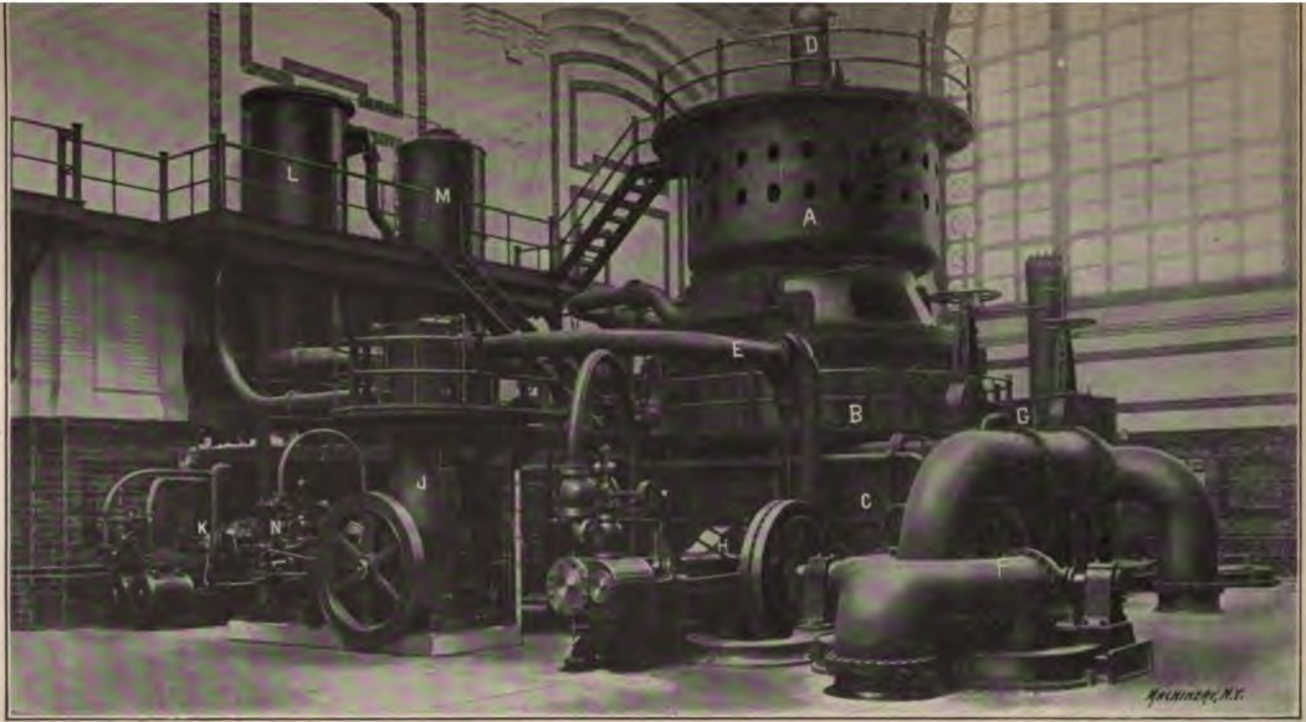


Fig. 5. Turbine and Auxiliaries at Station of Boston Edison Company.

that must be maintained under the step bearing of the turbine, etc., are all grouped about the turbine, in addition to the condenser auxiliaries. The lettered parts of the illustration are as follows: *A*, generator; *B*, turbine; *C*, condenser; *D*, governor; *E*, nozzles; *F*, circulating pump; *G*, accumulator for step bearing; *H*, engine to drive circulating pump; *J*, air pump; *K*, feed pump; *L*, heater; *M*, hot-well; *N*, air-pump engine.

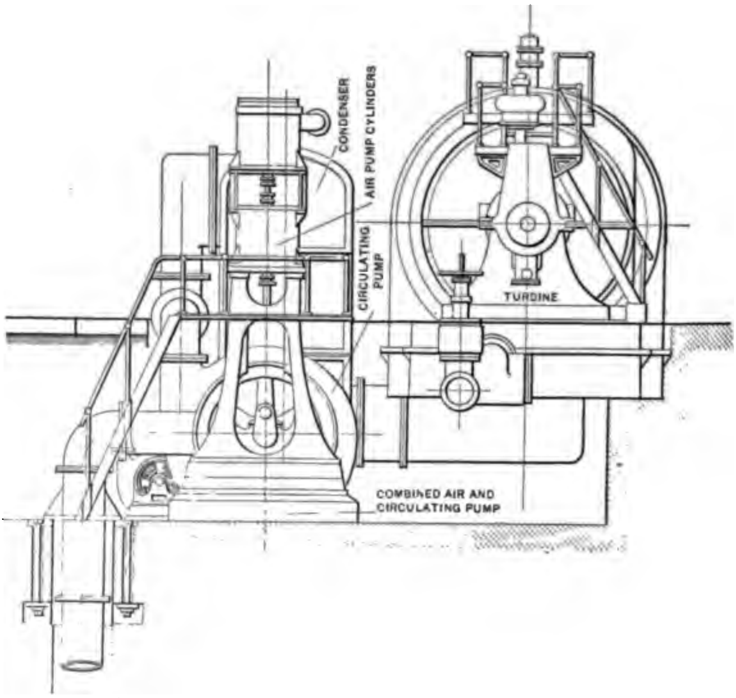


Fig. 6. End View of Turbine and Condenser Shown on Opposite Page.

Westinghouse-Parsons Turbine and Alberger Condenser.—Fig. 6 is an end view and Fig. 7 a plan of an Alberger surface condenser and apparatus applied to a 5,000 Kw. Westinghouse-Parsons turbine. The Alberger condenser is a counter-current condenser and does not require the use of a separate air cooler. The exhaust enters at the bottom and passes upward over the tubes. The cooling water enters at the top and passes downward,

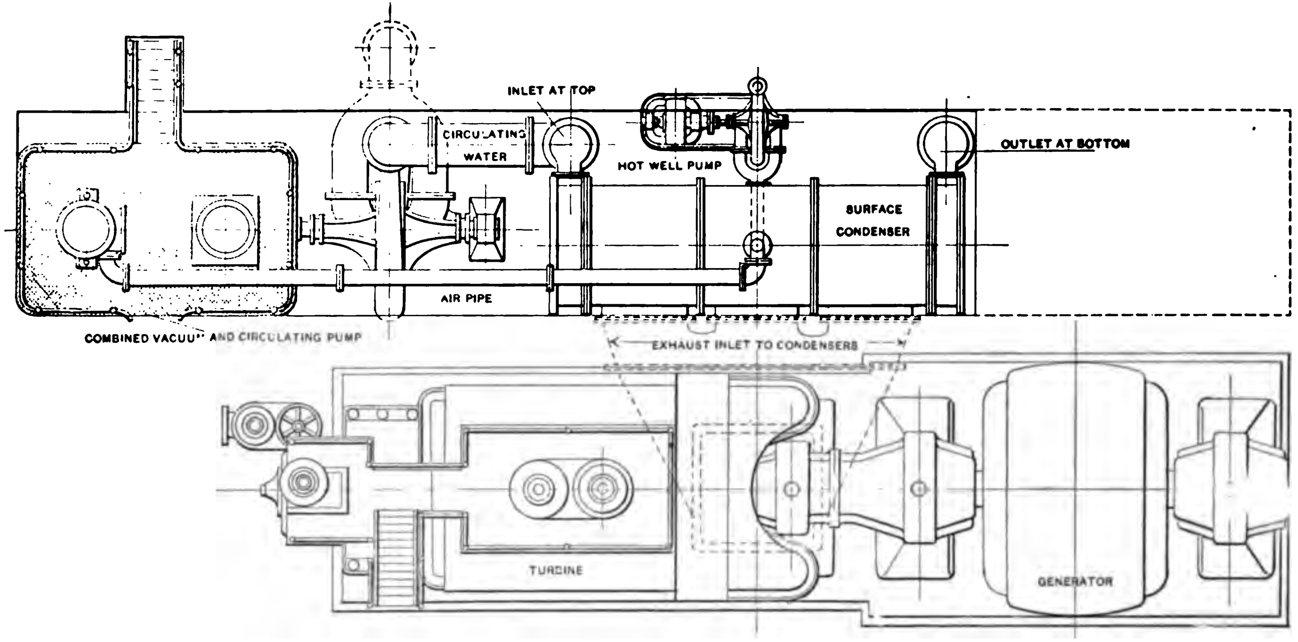


Fig. 7. Plan of Westinghouse-Parsons Turbine and Alberger Condenser, with Steam-Driven Auxiliaries.

back and forth, through the tubes. The air and vapors rising to the top of the condenser are therefore cooled by the in-coming cold water and the condensed steam which trickles down into the hot well is heated by the entering steam and a high hot-well temperature is thus maintained without difficulty. A two-stage vacuum pump is employed, one cylinder of which draws the air or vapor from the condenser and delivers it to the other cylinder which, in turn, forces it out against the pressure of the atmosphere. Space is saved and the apparatus simplified in this installation by using the same engine to drive both the circulating pump and the vacuum pump and it will be noted that the condenser and its pumps require just about the same area as the turbine itself, while by setting the condensing apparatus in a pit it rises only to the top of the turbine.

Condensing Apparatus Underneath the Turbine.—It is advocated by many that the proper place for the condenser and auxiliaries is on the same floor as the turbine, where they will be likely to receive better attention than if placed in a basement between the foundation piers or underneath the turbine. In view of the fact that a less massive foundation is required for a turbine than for an engine, however, it is possible to house the condensing apparatus perfectly under the turbine, as already explained in Chapter XVII. Such a plan, which in this instance is an exceedingly neat arrangement, has been followed in the installation at a power house of the Glasgow (Scotland) Corporation, Fig. 8. This is a Willans & Robinson-Parsons turbine. The condenser, circulating pump and Edwards air pump are clearly shown, the latter two being driven by electric motor. The lettered parts are: *A*, condenser; *B*, air pump; *C*, motor for pump; *D*, circulating pump; *E*, motor.

Parsons Vacuum Augmenter.—A novel arrangement for maintaining a high vacuum is the vacuum augmenter employed by Parsons in England with considerable success. He uses air pumps as shown in Fig. 9 placed below the level of the condenser and in addition to the usual pipe connection between the air pumps and the condenser there is a small pipe leading to the auxiliary condenser, generally having about one twentieth the cooling surface of the main condenser. In a contracted portion of

this auxiliary pipe is a steam nozzle which discharges a jet of steam that acts similar to the jet of an injector; this jet draws nearly all the residual air and vapor from the condenser and delivers it to the air pumps. The main pipe leading to the air pump is so curved as to form an air seal which prevents the air and vapor from returning to the condenser. With this arrangement there need be a vacuum in the air pumps of only about 26 inches, while the vacuum augmenter will increase the vacuum in the condenser to 27 or 28 inches. Mr. Parsons states that the quantity of steam required for the steam jet is about 1½ per cent of that

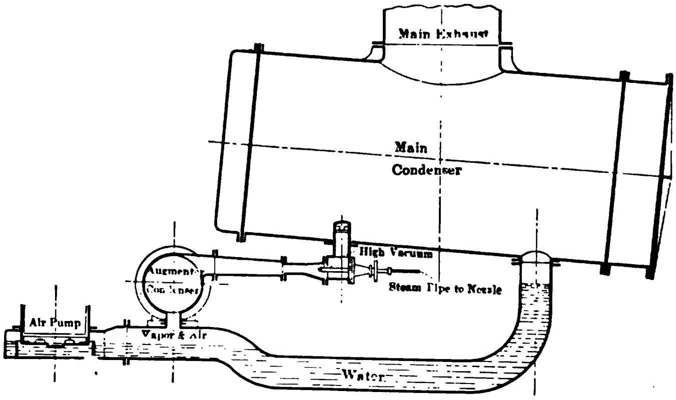


Fig. 9. Parsons Vacuum Augmenter.

Jet and Injector Condensers.

used by the turbine at full load, and this, together with the air extracted, is cooled by the auxiliary condenser.

Surface vs. Jet Condensers.—The surface condenser has come into extensive use with the steam turbine because the steam discharging from a turbine is entirely free from oil and if collected and condensed can be used over and over in the boilers. The feed water leaves the hot well of a surface condenser operating at high vacuum at nearly 100 degrees F., and passes through a heater where the temperature is raised still further by steam from the auxiliaries. In jet and injector condensers the condensed steam passes off with the injection water, which is at a temperature of

80 or 90 degrees, and when part of this is used for boiler feed there is a loss of some 10 or 20 heat units per pound, as compared with the surface condenser. This is so slight, however, that it does not pay to install a surface condenser and attending apparatus on the score of heat saved. The jet type is cheaper, simpler and works as well or better.

In localities where the available water supply contains sulphate of lime, acid, grease, or other harmful impurities, or where the cost of pure water is high, the surface condenser should probably be given the preference, though if it is merely the cost of the water that is at stake, the problem should be gone into very carefully before deciding.*

Injector Condenser.—A Bulkley injector condenser was installed by Geo. I. Rockwood in connection with a Westinghouse-Parsons turbine at Providence, R. I. The injection water is elevated into a vertical tank 30 inches square by 15 feet deep, in which the water level is maintained 6 inches below the water inlet nozzle of the condenser. The injection pipe takes the water from near the bottom of the tank. The air entrained with the water rises to the top of the tank and is largely eliminated from the injection water entering the condenser. The flow of the injection water through the throat of the condenser is what constitutes the air pump, and it is found to be the only air pump needed, since a vacuum of 28½ inches has been maintained, regardless of whether steam is passing through the turbine or not.

Jet Condenser.—A jet condenser, which is a modification of the

*In a paper before the A. S. M. E., December, 1904, Geo. I. Rockwood contends that the injector condenser would seem to bar out all other condenser systems in situations where the water is pure. He gives figures to show that it does not pay to install a surface condenser simply to save paying city rates for boiler feed water. His estimate for the cost of a high-vacuum surface condenser outfit is from $7 to $10 per kilowatt and of a jet or barometric condenser system from $5 to $6 per kilowatt.

In a paper before the American Railway Mechanical and Electrical Association, 1905, Fred N. Bushnell writes: "In cases where the cost of feed water is a material factor in the cost of power, or where it contains a large percentage of calcium or magnesium carbonate, or other scale-forming materials, there will be great advantage in using a surface condenser on account of the pure distilled water returned to the boilers, but where these conditions do not exist it will frequently be found practicable to use some simpler form of condensing apparatus such, for example, as the injector or barometric type of jet condensers. These types of condensers offer very great advantages over the surface condenser in the matter of lower first cost, space occupied, greater simplicity, and less cost of maintenance. Up to this time they have not been very generally used, but there seems to be no good reason why they should not work as satisfactorily in connection with steam turbines as with reciprocating engines."

injector condenser, is made by the Worthington company. As in the injector type, the condenser proper is placed about 30 feet above the hot well and the water falling, through the action of gravity, creates the vacuum. There is no contracted throat to this condenser, however, and the water is sprayed into the head, where it becomes intimately mingled with the steam before discharging through the vertical pipe. Fig. 10 shows a section of the condenser head. An air cooler and a dry vacuum pump are

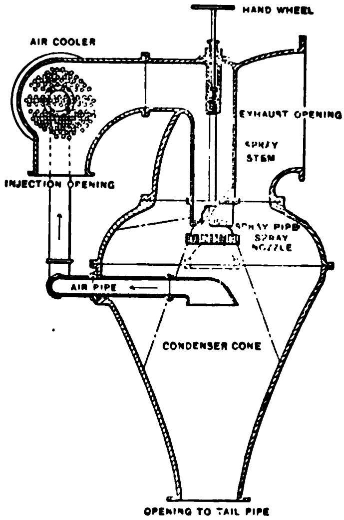


Fig. 10. Section of Worthington Jet Condenser.

employed, such as used with surface condensers, and any air that accumulates in the condenser head, where the steam is condensed by the spray of the water, is removed by the pump.

An interesting application of a jet condenser to a Parsons turbine is shown in Fig. 11. The plan is here adopted of substituting a centrifugal pump for the usual barometric column, enabling the condenser to be placed under the turbine. The exhaust steam is led through a pipe, *A,* and a gate valve, *B,* into the condensing chamber, *C;* and there, it is condensed by a jet and flows into the opening of a centrifugal pump, which is driven by a belt from the pulley on the extended shaft of the turbine.

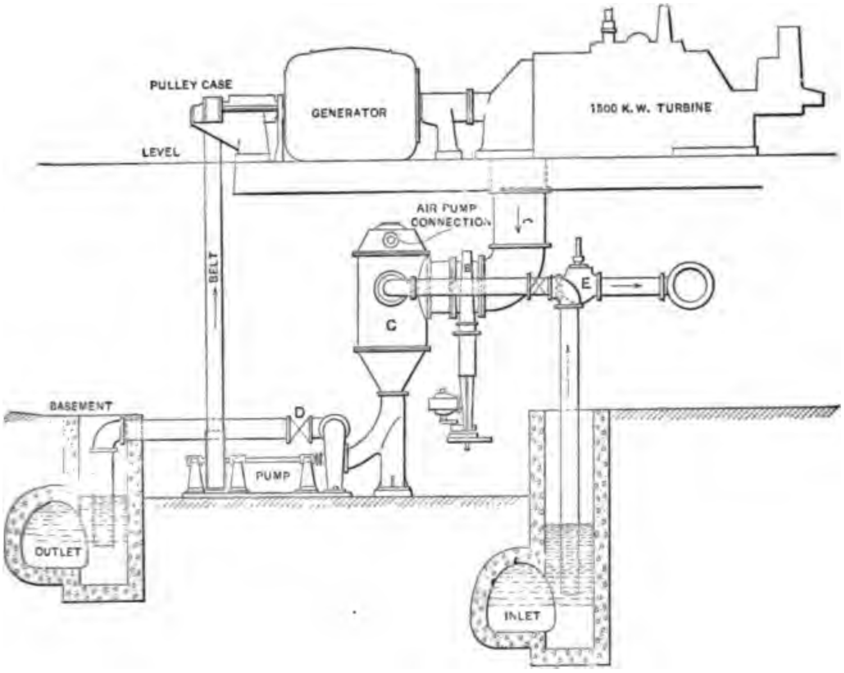


Fig. 11. Novel Arrangement of Jet Condenser with Centrifugal Pump.

There is a check, *D*, in the discharge from the pump, and this discharge is also sealed by the outgoing water. At the top of the condensing chamber is attached the usual dry air pump connection.

Data in Regard to the Performance of Condensers Operating Under High Vacuum.

Quantity of Cooling Water and Area of Condensing Surface.— An ordinary surface condenser giving 26 inches vacuum, with cooling water at 70 degrees, requires about 30 pounds of cooling water per pound of steam condensed and will condense about 10 pounds of steam per hour per square foot of surface. A surface condenser to operate with 28 inches vacuum will require more than twice this amount of cooling water, or say, 70 pounds per pound of steam (one manufacturer estimates as high as 85

pounds) and will condense about 5 pounds of steam per square foot of cooling surface. A common allowance in turbine work is 4 square feet of cooling surface per kilowatt. If the temperature of the cooling water is above 70 degrees the weight of water will have to be increased, sometimes very largely. A less quantity of injection water is required for jet or barometric condensers than for surface condensers. One manufacturer of the barometric type has furnished the author with the following figures for a 28-inch vacuum:—

Injection	at	40	deg.,	26	lb.	per	lb.	steam
"	"	50	"	29	"	"	"	"
"	"	60	"	35	"	"	"	"
"	"	70	"	50	"	"	"	"

A common allowance is 60 pounds per pound of steam for water at 70 degrees.

Tests on Condensers.—There are no data yet available, at least to the public, in regard to the performance of high-vacuum condensers, by which the relations between the several elements entering into the calculation of the quantity of cooling water, area of tube surface, etc., can be established, and results for the present must be more or less empirical.

Before calculations of condenser performance can be made, the initial and final temperatures of the steam and cooling water must be known. The following figures were given to the author at the works of the B. F. Goodrich Company, Akron, O., where Westinghouse-Parsons turbines are installed.

Test 1. Barometer 30 inches; vacuum 28.26; temperature steam in condenser 99.5; temperature hot well 99.05; initial temperature injection water 54 degrees; final temperature injection water 73 degrees.

Test 2. Barometer 30 inches; vacuum 28.61; temperature steam in condenser 100.5; temperature hot well 100.5; initial temperature injection water 57.9; final temperature injection water 72.5.

The data in regard to the quantity of water used and the power developed by the turbine during the tests were not sufficiently accurate to denote exact results, but they indicated about 60 pounds cooling water per pound of steam condensed.

The following results are from tests upon a surface condenser with Edwards air pump, in connection with a Curtis turbine which was running at a very light load:—

	Test 1.	Test 2.	Test 3.
Vacuum, inches,	27.89	27.64	27.92
Area cooling surface, sq. ft.,	2,700.	2,700.	2,700.
Temperatures, degrees F.:			
Steam in condenser,	103.	107.	102.
Hot well,	93.5	98.	93.5
Cooling water, initial,	75.	75.	75.
Cooling water, final,	96.	98.	93.
Weight steam per hour, lb.,	270.66	309.09	317.19
Weight cooling water per hour,	13,560.	14,094.9	19,634.2
Ratio, water to steam,	50.1	45.6	61.9

These three tests show a condition that is seldom met with in practice; viz., a final temperature of the cooling water equal to or higher than the hot-well temperature. This was attained because of the small quantity of steam condensed per square foot of cooling surface. Under ordinary conditions the final temperature of the cooling water will be from 10 to 25 degrees below the hot-well temperature. In counter-current condensers the temperature will be higher than in the parallel flow type. The final temperature is also dependent upon the quantity of cooling water forced through the condenser tubes and upon the area of the tube surface.

Condenser Calculations.—The following simple example shows the method of calculating the weight of cooling or injection water when the temperatures are known. No allowance is here made for the efficiency of the condenser, which must be determined by experiment, but the example will explain why it is so difficult to maintain a high vacuum.

Example:—The temperature of the steam in a condenser at 28 inches vacuum is about 100 degrees, and its total heat per pound is found from the steam tables to be 1,112 units. By the use of a dry vacuum pump it is possible to secure a hot well temperature of 98 degrees, and the heat in each pound of condensed steam is therefore 98 — 32 = 66 units. Hence there are 1,112 —

66 = 1,046 heat units given up to the cooling water per pound of steam condensed.

With cooling water at 70 degrees initial and 100 degrees final temperature (the latter equal to the temperature of the condensed steam), the water will have risen 30 degrees and gained 30 heat units per pound.

But the heat lost by the steam was 1,046 heat units per pound. Hence, 1,046÷30=34.9 for the ratio of cooling water to condensed steam.

In practice no condenser can work with so high an efficiency as in this case, where the cooling water takes up all the heat of condensation of the steam and leaves the condenser with the temperature of the condensed steam. Under normal conditions the final temperature of the cooling water will range from 10 to 25 degrees below the temperature of the condensed steam, as previously stated. Let us assume it to be 15 degrees below. Then, with cooling water at 70 degrees initial and 85 degrees final temperature, we have:—

Heat units absorbed per pound of water=85—70=15; and 1,046÷15=69.7. for the ratio of cooling water to condensed steam, or double what it was before.

In winter time, when the initial temperature of the cooling water would be about 40 degrees, we should have, assuming a final temperature of 85 degrees, 85—40=45 and 1,046÷45=23.2 for the ratio. As a matter of fact it would be more likely that the plant would be operated with as much cooling water in winter as in summer to derive the benefit of the higher vacuum that would be secured; and in this case the final temperature would drop to, say 25 degrees below the temperature of the condensed steam, as was the case in the tests upon the Goodrich plant previously quoted.

On the other hand, if cooling towers were employed and the temperature of the cooling water rose to 80 degrees or more in the summer time, it is evident that the square feet of cooling surface of the condenser should be on a liberal basis, in order to secure as high a final temperature of the cooling water as possible; otherwise the quantity of water to be circulated might be almost prohibitive in amount. Even under favorable conditions the prob-

lem of maintaining a high vacuum in connection with a cooling tower is doubtful of solution.

*Power to Operate Auxiliaries.**—The plant of the Citizens' Light, Heat and Power Company, Johnstown, Pa., is partially equipped with turbines and Weiss jet condensers of the barometric type. The condenser auxiliaries are driven from a single steam cylinder—that of the rotative air pump—and by indicating this cylinder at normal speeds the total power in-put was obtained. The results were reduced to percentages of the turbine output and showed that at less than one quarter load the total power consumption was less than five per cent of the sustained output, and that it progressively decreased to 2½ per cent at full load.†

J. R. Bibbins states in a paper, "Steam Turbine Power Plants," that observations on the 2,000 Kw. turbine at the St. Louis Exposition indicated the following results: At full load the total power in-put to the auxiliaries was 7 per cent; approximately three fourths required for the circulating pump, two ninths for the air pump, and one fourteenth for the hot well pump. This 7 per cent represents gross power and includes transformer and motor losses. The cooling water, however, was exceptionally warm, being fre-

*In the *Journal of Electricity, Power and Gas* for March, 1905, was the first of several extended articles by Charles C. Moore & Co., incorporated engineers at San Francisco, Cal., upon the estimated power for operating condenser auxiliaries under above conditions. The following quotation is taken from the introduction: "The quantity of circulating water required for high-vacuum condensing plants must be increased from the old standard of from 25 to 30 pounds to from 40 to 60 pounds of water per pound of steam condensed for moderate temperatures, and from 60 to 100 pounds of water per pound of steam when used at the higher temperatures common to cooling tower practice. In cases of excessive head or quantity of circulating water the bulk of the power required by auxiliaries is due to the circulating pump. The range of power for this purpose varies so widely that the older method of assuming a given type of plant requiring 5, 10, or 15 per cent of the total steam consumption to drive auxiliaries is entirely in error without an accompanying statement defining conditions under which circulating water is pumped. The power to drive the air pump is dependent somewhat upon the vacuum, but particularly upon the air leakage into the condensing system. It was for some time assumed that the work of the air pump corresponded to removing the air which entered the boiler in solution in feed water. As a matter of fact, handling the air in solution is the smallest portion of work done by air pump, the leakage through piping, pipe joints, pores of castings, stuffing boxes, etc., imposing the greatest duty, the total quantity of air to be handled ranging from ten or fifteen to thirty or forty times the air dissolved in ordinary water. The actual power to drive the air pump should in good practice be less than .00018 indicated horse-power in the air pump cylinder per pound of exhaust steam per hour. As the amount of power necessary to drive the air pump is a comparatively small portion of the total power for auxiliaries a slight error in this quantity will not largely affect the final result."

†*Power*, February, 1905.

quently 85 degrees, which made it difficult to obtain the high vacuum desired.*

The following tests upon the auxiliaries of the 5,000 Kw. unit of the Boston Edison Company, shown in Fig. 5, were reported in the report of the turbine committee of the National Electric Light Association for 1905:—

	Test 1.	Test 2.	Test 3.
Kilowatts on turbine,	2,713.	3,410.	4,758.
Vacuum,	28.4	28.7	28.6
Barometer,	29.53	29.95	29.96
	Horse-power Developed.		
Boiler feed pump,	13.9	23.7	27.4
Circulating pump,	69.1	69.1	69.1
Dry vacuum pump,	24.3	23.2	23.8
Step bearing pump,	6.4	5.8	5.6
Wet vacuum pump,	8.6	9.2	9.8
Totals,	122.3	131.	135.7
Per cent power of auxiliaries to power of turbine,	3.4	2.9	2.1
Per cent water used by auxiliaries to that used by turbine,	8.4	7.4	5.7

*American Street Railway Association, 1904.

CHAPTER XX

THE STATUS OF THE MARINE TURBINE.

Early History.—Most of the turbines applied to the propulsion of vessels have been of the Parsons type, although some work of this character has been done both by Rateau and Curtis. In 1894 the Parsons Marine Steam Turbine Company, Ltd., Wallsend-on-Tyne, England, was formed and the experimental boat *Turbinia* constructed. Her dimensions were 100 feet beam, 3 feet draft and 44 tons displacement. There were three separate turbines—a high-, an intermediate-, and a low-pressure, each driving a screw shaft and on each shaft were keyed three propellers of small diameter. The turbines were rated at 2,000 horse-power and the boat attained a speed of over 34 knots.

Various other high speed boats were built during the next five or six years. Two of these, the *Viper* and *Cobra,* high speed torpedo boat destroyers, were lost at sea and turbine propulsion received a serious setback. An organization was finally effected, however, which included the shipbuilding firm of Messrs. Denny, the Hon. Charles A. Parsons and Capt. John Williamson, which resulted in the first turbine steamer, *King Edward,* in 1901, for service on the Clyde.

The First Turbine Steamer.—The *King Edward* is a boat 250 feet long, 30 feet beam with 6 feet draft. The arrangement of the machinery is practically the same as has been used in all the more recent vessels, including the ocean liners, fitted with Parsons turbines. There are three separate turbines driving three screw shafts. The high-pressure turbine is placed on the center shaft and the two low-pressure turbines each drive one of the outer shafts. Inside the exhaust ends of each of the latter are placed the two astern turbines which rotate as one piece with the low-pressure motors and when in operation reverse the direction of rotation of the low-pressure motors and outside shafts.

In ordinary going ahead steam from the boilers is admitted to the high-pressure turbine and after expanding about 5 times passes to the low-pressure turbines and is again expanded in

them about 25 times and then passes to the condensers, the total expansion ratio being about 125 as compared with from 8 to 16 usual in triple expansion reciprocating engines of the marine type. At 20 knots the speed of the center shaft is 700 and of the two outer shafts 1,000 per minute.

When maneuvering in or out of harbor the outer shafts only are used and the steam is admitted by suitable valves directly into the low-pressure motors or into the reversing motors, for going ahead or astern. The high-pressure turbine under these circumstances revolves idly, its steam admission valve being closed and its connection with the low-pressure turbines being also closed by non-return valves.

Later Turbine Boats.—Following the *King Edward,* and a later boat for the same line, the *Queen Alexandria,* has come a long list of other turbine vessels, notably a fleet of 18 cross-Channel boats built or building, to ply between Dover and Calais. Again, on the Heysham line running between Great Britain and Ireland, turbine vessels have been in successful operation. In 1904 the third-class turbine cruiser *Amethyst* was built for the British Admiralty. She is 360 feet in length and of 3,000 tons displacement. Three other engine-driven cruisers of the same size were built simultaneously, one of which, the *Topaz,* was selected for a series of competitive trials with the *Amethyst.*

The contract speed of the vessels was 21¾ knots, and the results showed that at all speeds above 14½ knots the turbine vessel was the more economical, at 18 knots the turbine was 15 per cent more economical, at 20½ knots 31 per cent, at 22.1 knots 36 per cent, and at full power in each vessel the *Amethyst* showed 42 per cent more power than required by contract on the coal allowed; while the *Amethyst* reached 23.6 knots on the specified coal and the *Topaz* only 22.1 knots. In other words, the *Amethyst* has a radius of action at 20 knots speed of 3,600 nautical miles, while her sister vessels with ordinary engines can only steam 2,000 miles at the same speed.

The success of the *Amethyst* led British naval constructors to advocate the turbine for larger vessels and the activity of the admiralty following the Russo-Japanese war culminated in the construction of the powerful battleship *Dreadnought.* This ship,

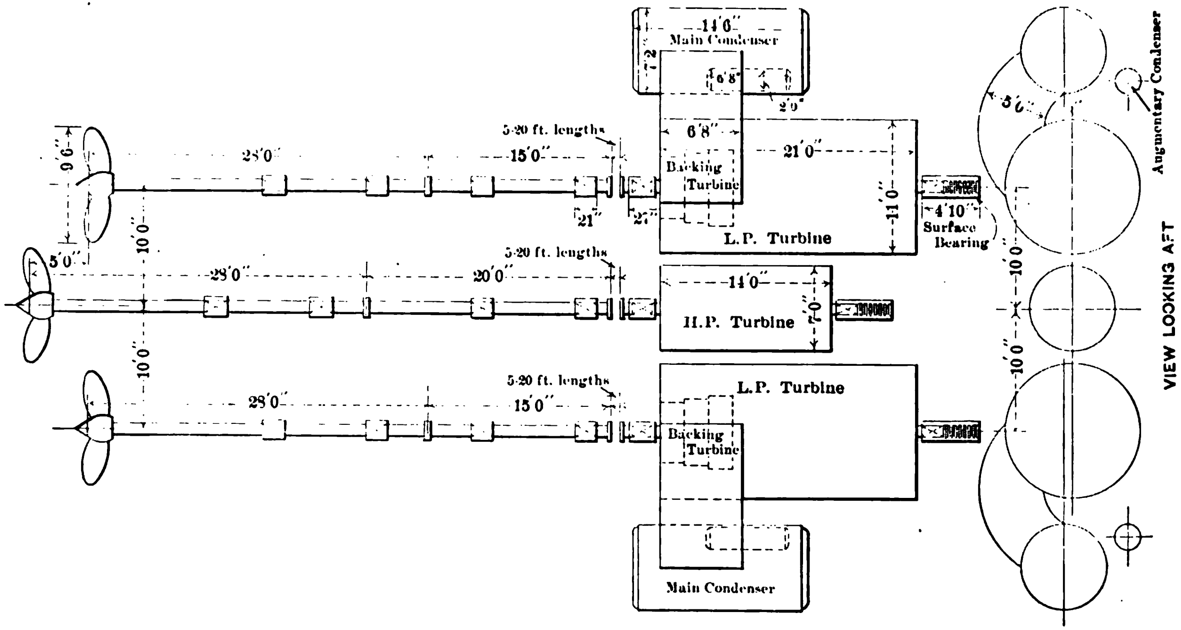


Fig. 1. Arrangement of Turbine and Propellers of the Atlantic Liners "Victorian" and "Virginian."

which is not only larger and carries a heavier armament than any battleship afloat, is remarkable because it is the first battleship to be driven by turbines. Under trial the turbines developed 28,000 horse-power and propelled the vessel at an average speed of 21½ knots during a trial of eight hours, acquiring a maximum speed of 22¼ knots. The turbines are so free from vibration that the ship makes the steadiest possible gun platform for a floating battery.

Atlantic Liners Fitted with Turbines.

Description of the Steamer Victorian.—The Allan liners *Virginian* and *Victorian* started to ply between Liverpool and Canada in the summer of 1905. In April, 1905, Commander A. D. Canaga, United States Navy, was detailed to make the trip to Europe and return on the turbine steamer *Victorian* and report the results of his observation to the department.* There unfortunately is no similar vessel of the same line propelled by reciprocating engines with which a direct comparison can be made, but certain points brought out by Commander Canaga will be of interest. Figs. 1, 2, and 3 are reproduced from his report showing the arrangement of turbines in this vessel which is like that usually adopted for the Parsons' apparatus, and is practically the same as already described in connection with the *King Edward.* The steam from the boilers is led into the engine room through two 12-inch pipes, uniting in the throttle valve at the working platform. From the throttle valve steam is led through two 12-inch pipes to the high-pressure turbine. When in free route the steam is passed through the high-pressure turbine where it spreads, half going to the starboard and half to the port turbine through the receiver pipes, and thence through exhaust pipes to the main condensers. In maneuvering, the main throttle is closed and steam admitted to the maneuvering valves, Fig. 3, one for each low-pressure turbine. These are simple slide valves which when placed at the upper end of their stroke admit live steam to the forward end of the low-pressure turbine, when at the bottom of their stroke admit live steam to the backing turbine, and when

**Journal of the American Society of Naval Engineers,* August, 1905.

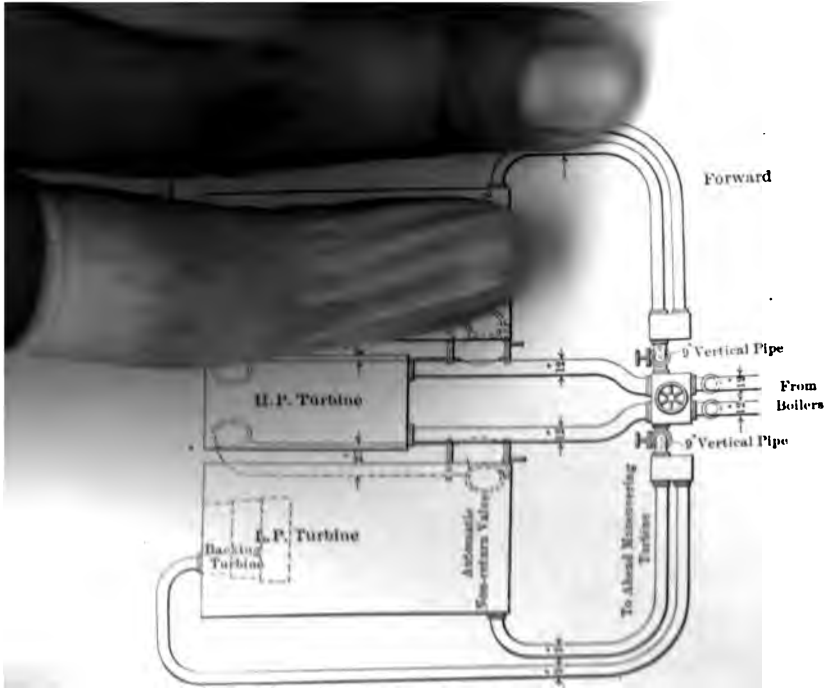


Fig. 2 Showing Pipe Connections of Marine Turbines.

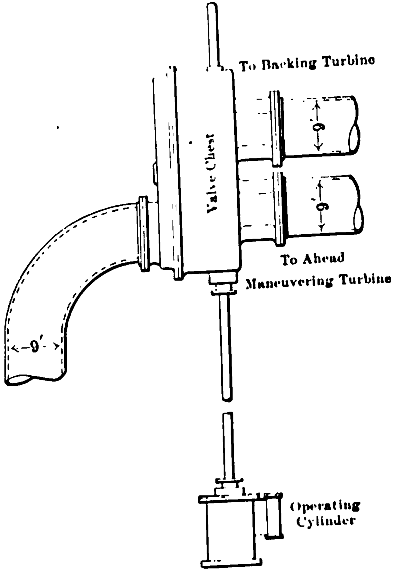


Fig 3. Arrangement of Operating Valve.

in mid-position shut the steam from both the ahead and backing turbines.

To prevent the steam blowing off into the high-pressure turbine when maneuvering, non-return valves are fitted in the receiver pipes between the high-pressure and low-pressure turbines, as shown in Fig. 2. These valves are automatic, opening or closing as the live steam is admitted to the high-pressure or low-pressure turbines.

Comments on the Operation of the Victorian.—At the forward end of each turbine shaft is fitted a safety governor, which in case of accident closes the main throttle valve. These governors serve another purpose, also, by indicating whether the turbines are at rest or in motion, since from the working platform they are the only visible moving parts. The commander reports that the turbines are easily and quickly handled and that the minor mishaps and annoyances met with in reciprocating engines are absent. He notes a pleasing absence of vibration and of racing in high seas. Against this immunity from racing, however, must be set the lack of holding power of the small screws with which turbine vessels must be equipped. It was observed that the influence of head winds and heavy seas reduced the vessel's speed considerably more than would have been the case with the large propellers used with reciprocating engines.

Attention is called to the fact that heating of bearings is a more serious matter with turbine machinery than with reciprocating engines as any unusual wear would cause interference between the rotating and stationary blades, and on one of the trips the *Victorian* was delayed about 29 hours owing to some grit that got into one of the bearings and necessitated overhauling, after which, however, there was no trouble. There was also considerable difficulty from priming of the boilers, but with consequences less serious than in the case of reciprocating engines.

Turbine Boats of the Cunard Line.

The *Carmania** was built and equipped by Messrs. John Brown & Co., Ltd., Clydebank. This vessel is one of the seven or eight

*Taken in part from *London Engineering*, December 1, 1905.

largest of the world's ships, having the following dimensions: length between perpendiculars, 650 feet; length over all, 672 feet, 6 inches; breadth, moulded, 72 feet; depth, moulded, 52 feet; gross register tonnage, 19,524 tons; draft, in working condition, 33 feet, 3¼ inches; displacement at this draft, 30,918 tons. She was designed for a speed of 19 knots.

The Turbines are of the Parsons type arranged in the usual manner, with the high-pressure turbine on the center screw shaft and with one low-pressure turbine on each of the outside shafts. The arrangement of the turbines and auxiliaries is shown in Figs. 4 and 5. In normal working conditions the regulating valves admit steam to the high-pressure turbine and the steam passes through it to the low-pressure turbines and thence to the condenser. For maneuvering purposes a large, non-return valve, worked by a steam and hydraulic engine controlled from the starting platform, closes the connection between the high-pressure and each low-pressure turbine. This valve will close automatically as soon as a prescribed pressure is obtained in the low-pressure casing. Each of the low-pressure turbines is then manipulated as an independent unit. Special maneuvering valves are fitted to each of the low-pressure and astern turbines, which allow of steam being admitted to either the ahead or astern turbines by a single movement of a hand lever. The turbine is fitted with a governor which operates when there is any marked increase in the number of revolutions. Oil is supplied to the bearings under considerable pressure. It then flows to a cooling tank fitted with copper coils through which water is circulated and after passing through a system of oil filters is again delivered to the bearings.

The Turbine Glands.—The gland for the shaft passing through the end of the turbine is rendered steam-tight by an improved design, Fig. 6. In the Parsons marine turbine of smaller sizes the leakage of steam from the high-pressure turbine and the ingress of air into the low-pressure turbine has been prevented by the Parsons ring-and-groove type of gland. This consists of a series of grooves turned in the spindles into which bronze rings are fitted, the whole rotating in a truly bored cylindrical gland. When the difference in pressure between the inside of the turbine and the atmosphere is great the side pressure of these rings is very con-

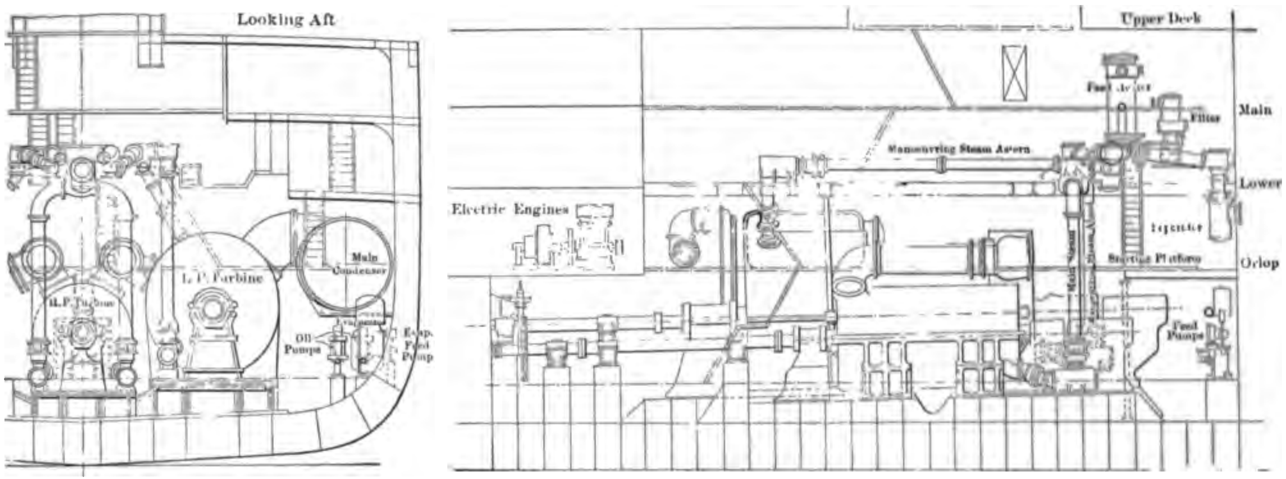


Fig. 4. Elevation of Turbines of the "Carmania."

siderable and in order to distribute this pressure as equally as possible the rings are arranged in groups and the pressure to each group is graded by suitable connections (Fig. 7).

As a result of experiment with specially constructed apparatus it was found the regular construction would not answer for glands of the large diameter required for the *Carmania.* The speeds and pressures were so high that the rings wore rapidly. Finally radial fins, as in Fig. 8, were used in connection with a row of rings and grooves. The action of these fins is to alternately wire-draw and expand the steam, each pair constituting an expansion stage, thus reducing its pressure as it travels outward. The actual gland was fitted at each end of both the high- and low-pressure turbines, as illustrated in Fig. 6, with four rings, *K,* at the outer end. The small amount of steam which is allowed to leak past them for the purpose of lubrication collects in pocket *G,* whence it is led by the pipe *H* to the auxiliary condenser or exhaust tank. In the case of the high-pressure turbine, where the radial fins do not sufficiently reduce the pressure of the escaping steam, the pocket *O* is connected to an expansion row in the low-pressure turbine.

Comparison with the Coronia.—The area occupied by the turbines and auxiliaries is practically the same as required for the quadruple-expansion reciprocating engines of the sister ship *Coronia,* built sometime previously. The required head room is less, but no advantage is taken of this, as the space above the engine room was left open for light and air. There is a saving in weight of about five per cent. The boiler pressure in the *Coronia* is 210 pounds and in the *Carmania* 195 pounds per square inch. The turbines take steam at an initial pressure of 150 pounds as against 200 pounds in the quadruple engines. The cooling surface of the condensers is increased in the *Carmania* about 20 per cent, the capacity of the centrifugal pumps is about double, and the weight of circulating water is from 50 to 60 times the weight of feed water as compared with a ratio of 25 or 30 times in the *Coronia's* installation.

The *Lusitania* and *Mauritania,* of the Cunard Line, which are expected to become the queens of the sea, are turbine vessels designed to maintain a minimum speed of 24 to 25 knots. The dimensions of the *Lusitania,* which is more nearly completed than

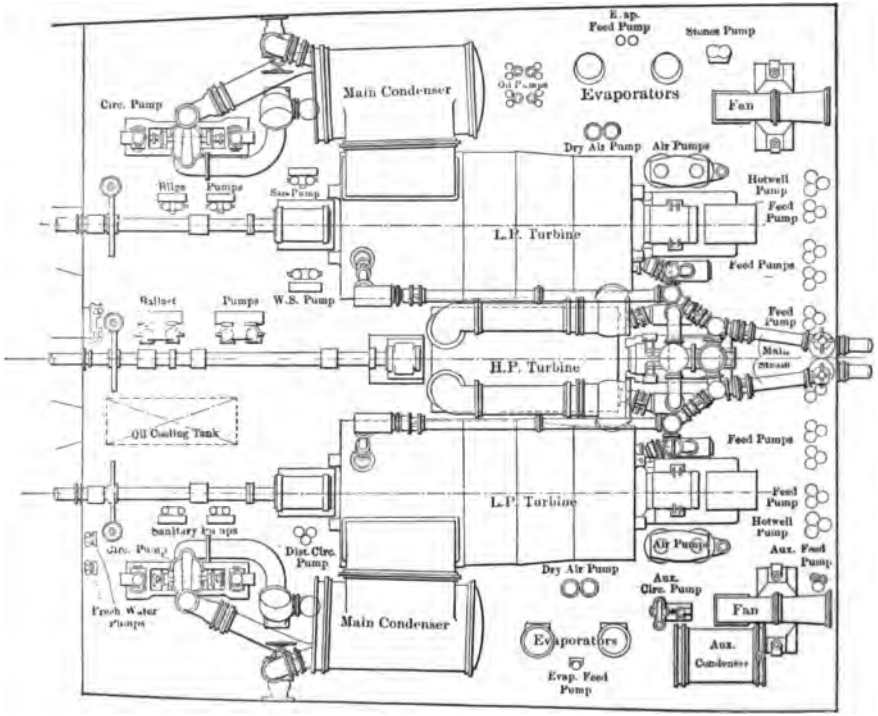
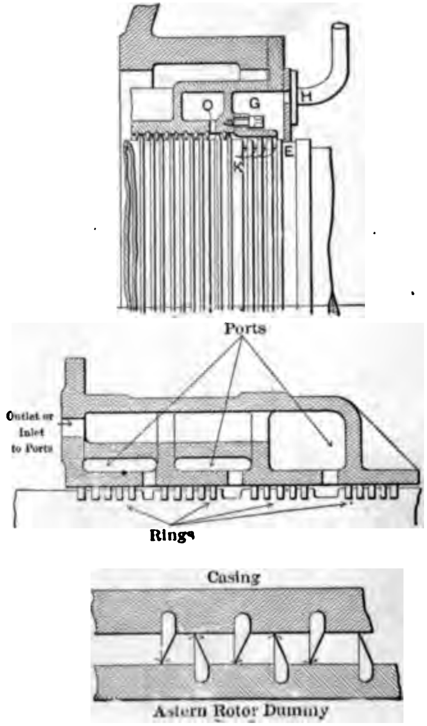


Fig 5. Plan of Turbines of the "Carmania."



Figs. 6, 7 and 8. Details of Glands.

the sister ship, are a length of 785 feet over all, extreme breadth of 88 feet and a depth of 60½ feet. There are 25 Scotch boilers, all but two of which are double-ended, and carrying a pressure of about 200 pounds per square inch. The turbines are to develop 70,000 horse-power, an increase of 70 per cent over that developed by the largest and fastest vessel previously constructed. The turbines are divided into four units, each of which has its own propeller shaft and propeller. The two outer shafts carry the high-pressure turbines and the inner shafts the low-pressure and backing turbines, making six in all, four for forward motion and two for backing. The two high-pressure propellers are located about 80 feet forward of the low-pressure propellers and are carried by long, tapering tubes, which support them at quite a distance from the sides of the hull.

The longest blades on the low-pressure end are 20 inches in length and the peripheral speed of the rotating blades ranges from 100 to 150 feet per second. The blades are bound together with brass and copper wire soldered. The casings and blades for one of the low-pressure turbines weigh 450 tons and are made up of six sections. The rotors of the high-pressure turbines are about seven feet in diameter and 25 feet long, while those of the low-pressure turbines are somewhat larger. It is stated that such accuracy is being attained in the construction of the turbines that it is expected to operate them successfully with a clearance between the blade tips and the casing, or the rotor, of only $\frac{1}{20}$ inch.

The construction of these great vessels was induced by the success of the recent new and fast German vessels of the Hamburg-American line in wresting away the supremacy of the seas. The fastest of these, the *Kaiser Wilhelm II.*, has attained a speed of 23½ knots. As a matter of pride it was desired to restore to the English merchant marine the distinction of having the "largest and fastest" afloat, and as a matter of safety to the government the Admiralty wanted more fast merchant vessels that could be drafted into the service in time of war. To obtain the increase of speed from the 23½ knots of the fastest German vessel to the 25 knots called for in the turbine ships, a tremendous increase in engine power was necessary, amounting to 70 per cent, as stated

above, and it is doubtful if such great power could be successfully generated in the hold of a ship by means of reciprocating engines.

Both the machinery and hull of the *Lusitania* are from the Clydebank Works of Messrs. John Brown & Co., Ltd.

Comparison Between Turbines and Reciprocating Engines.

The best opportunity for comparing the performance of vessels fitted with turbines and engines has been afforded by the Midland Railway Company's four boats of the Heysham Line of Great Britain. Of these, the *Londonderry* and *Manxman* have turbines and the *Antrim* and *Donegal* reciprocating engines. The *Londonderry, Antrim* and *Donegal* have the following dimensions: Length 330 feet, breadth 42 feet, depth 25 feet, 6 inches. The *Manxman* is of the same length and depth, but has a breadth of 43 feet. The turbine boat *Londonderry* carries 150 pounds boiler pressure and the others 200 pounds pressure. The engines of the *Antrim* and *Donegal* are of the triple-expansion type, differing only in details, and drive a single, three-bladed propeller. The turbines of the *Manxman* were designed for 25 per cent more power than those of the *Londonderry,* but are of similar construction and drive three three-bladed screws after the usual manner. All the boats have high-grade condensing apparatus, but the *Manxman* has in addition a Parsons vacuum augmenter, for producing a high vacuum.

Official Trials of Heysham Line Boats.—The results of the official trials showed the two boats with reciprocating engines to be on a par in economy and to use practically the same amount of feed water under like conditions. At speeds of 19 to 20 knots, however, which is the working speed of all the vessels in service, the water consumption of the turbine steamer *Londonderry* was 8 per cent less and of the *Manxman* 14 per cent less than of the *Antrim* and *Donegal,* while throughout a speed range from 1 to 20 knots the turbine boats showed superior economy. Speed trials were run between the two turbine boats and the *Antrim* and the *Londonderry* proved about one knot faster and the *Manxman* from one to two knots faster than the *Antrim,* under like conditions.

Results based on the Log Books.*—Later, comparisons were instituted between the turbine steamers and those with reciprocating engines, based on the log books in which the daily records were kept while the boats were in regular service. During a part of this comparative period the *Manxman* was not on the same route as the other vessels, so that she could not be consistently compared with the *Antrim* and *Donegal* during the entire time. Also, the high-pressure turbine of the *Londonderry* was partially wrecked, owing to the blades of the rotating drum coming in contact with the stationary blades, so that the records from this steamer were interrupted for three months. Valuable comparative figures were secured, however, and are summarized in table below. The regular route was between Heysham and Belfast, one vessel plying each way every night except Sunday. The comparisons are made in each case between vessels running in opposite directions on the same days and the table gives the weight of coal each vessel consumed on a given number of trips, exclusive of that burned when in port, which of course does not affect the performance of the propelling machinery.

TABLE SHOWING RESULTS OBTAINED BY STEAMERS RUNNING SIMULTANEOUSLY, BUT IN OPPOSITE DIRECTIONS.

	Reciprocating Engines	Turbines.
	Antrim.	Londonderry.
Number of Trips	48	48
Average Coal per Trip, tons	35.6	35.8
Average Speed in knots	19.7	19.5
	Donegal.	Londonderry.
Number of Trips	42	42
Average Coal per Trip, tons	36	36.9
Average Speed in knots	19.2	19.8
	Antrim.	Manxman.
Number of Trips	29	29
Average Coal per Trip, tons	38.6	38.6
Average Speed in knots	19.5	20.3
	Donegal.	Manxman.
Number of Trips	39	39
Average Coal per Trip, tons	38.7	40.2
Average Speed in knots	19.3	20.3

An economy in the turbine steamers is the small amount of oil required, only five gallons being used per trip, and the dispensing

*Reported in *London Engineering*.

with the services of two oilers usually required. There is the absence of vibration, but there is also the inferiority in maneuvering from rest in narrow waters. Experiments made during the trial trips of these turbine boats indicate that they may be brought to rest from full speed in about 1½ minutes. This is a good result, but experience has shown the inadequacy of the backing power from rest, when the side propellers are less efficient.

Advantages and Disadvantages of the Marine Turbine.—The chief advantages, on the evidence already given, appear to be absence of vibration, making the turbine boat a much pleasanter passenger craft; greater speed with the same amount of coal or less coal at the same speed, when running at or near normal speeds; ease of manipulation; slight saving in weight; less oil; less attendance; less liability of racing.

The chief disadvantages are the lack of holding power of the small screws; the diminished power of the reversing turbines; and the poorer economy at low speeds.

Cavitation.—The size of the propellers on turbine vessels is limited by trouble experienced through cavitation. When the speed of a propeller blade exceeds a certain amount, depending upon the type, the head or pressure is not sufficient to keep up the supply of water to the propeller and a partial vacuum is formed back of the blade, in consequence of which the efficiency of the propeller drops off. This action is known as cavitation. Marine turbines are made larger in diameter for a given power than land turbines used to drive electric generators, and in this way the speed of rotation is reduced somewhat. The speed still remains so high, however, that the size of propeller must be reduced to avoid cavitation; and then, to secure a sufficiently low thrust per square inch of propeller area the blades are made wide. A wide-bladed propeller has usually been considered inefficient, due, probably, to the increased friction that such a propeller would have if of the usual large diameter. This objection is not so serious with small turbine propellers, however, which show a reasonably high efficiency.

APPENDIX

Figs. 1 and 2 on the succeeding pages show the kinetic energy of a steam jet in foot-pounds, and Figs. 3 and 4 the velocity of a steam jet in feet per second, under the assumption that the steam is initially dry and that it attains its velocity as the result of adiabatic expansion. The pressures indicated on each curve are initial pressures and the figures at the left of the diagrams are final pressures. The values at the bottoms of the diagrams give the energy or velocity, as the case may be, of dry, saturated steam expanding from a given initial to a given final pressure. For example, steam at a pressure of 165 pounds absolute and discharging from a nozzle at 95 pounds absolute would acquire a velocity of about 1,500 feet per second, if there were no losses of energy; and the jet would develop approximately 35,000 foot-pounds of energy per pound of steam flowing. If one pound discharged per minute, the jet would develop a little over one horse-power.

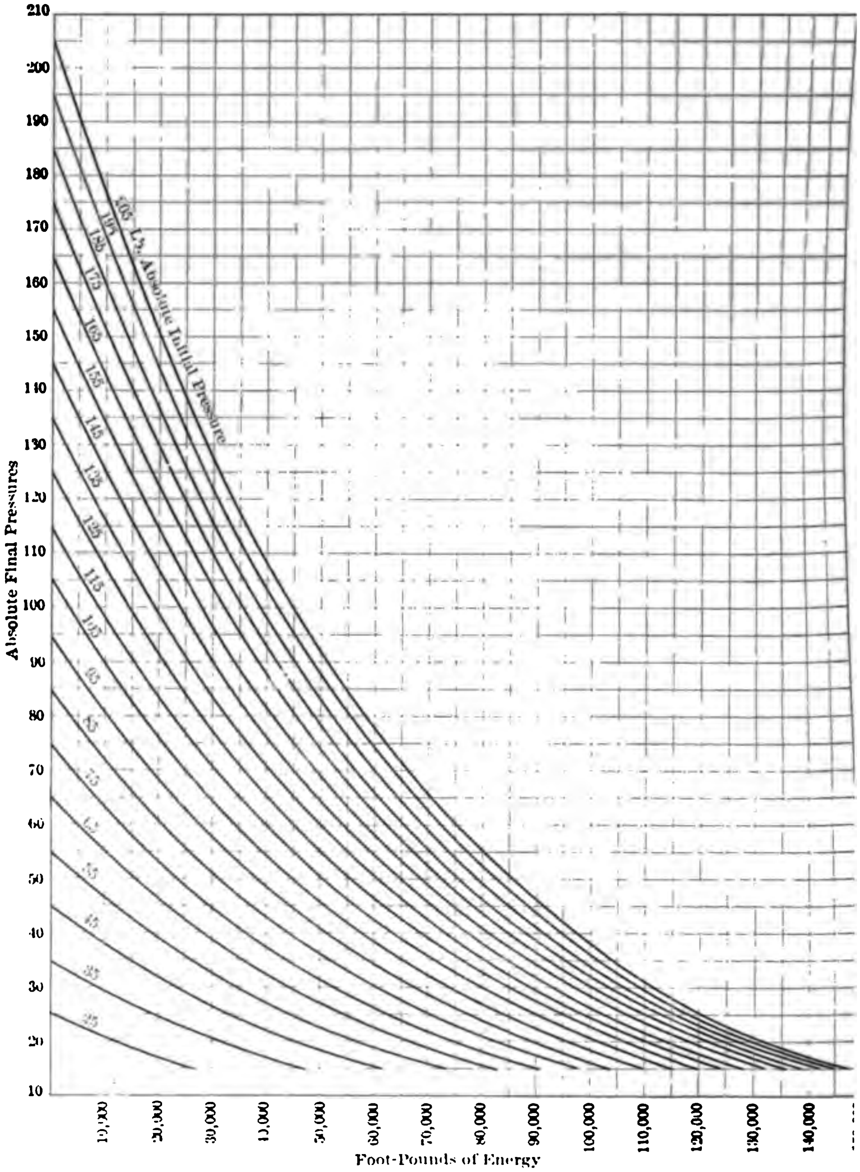


Fig. 1. Energy of a Steam Jet in Foot-pounds, when the Steam Expands Adiabatically between different Initial and Final Pressures.

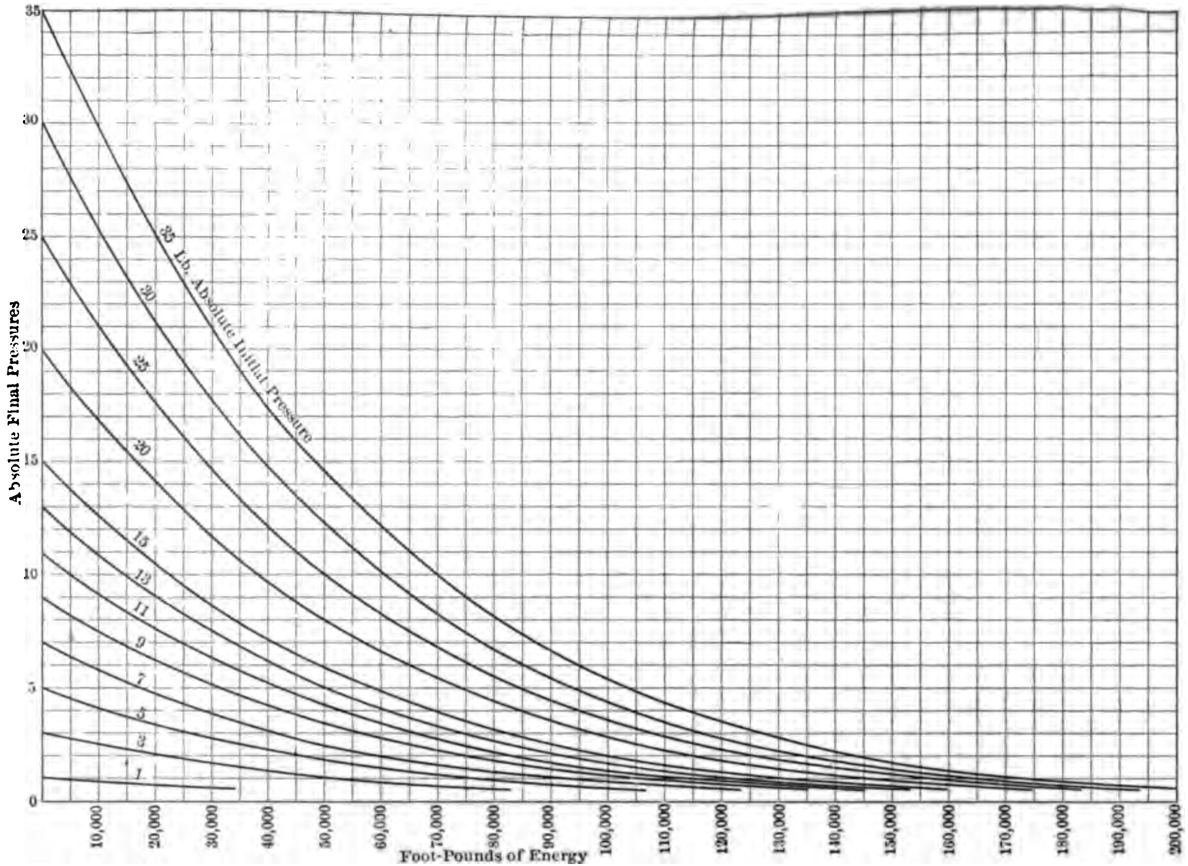


Fig. 2. Energy of a Steam Jet in Foot-pounds.

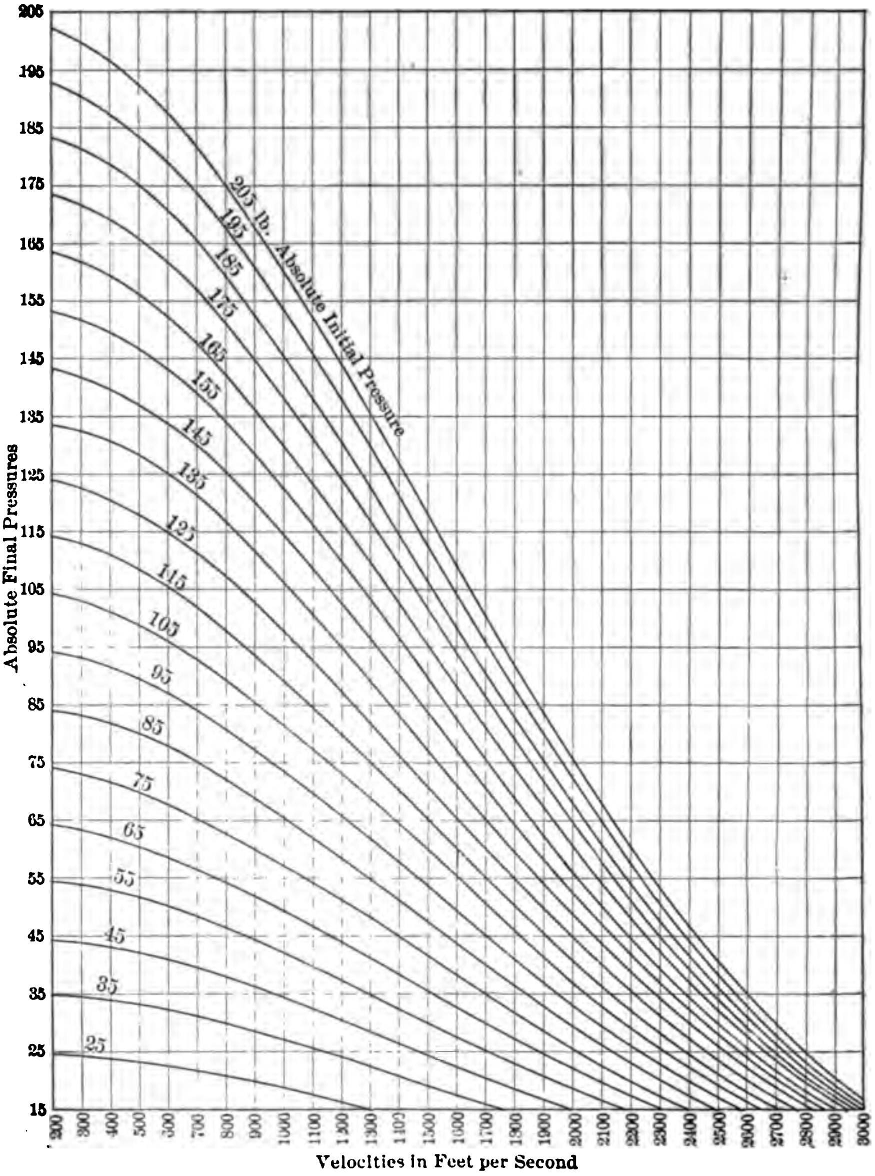


Fig. 3. Velocity of a Steam Jet in Feet per Second, when Steam Expands Adiabatically between different Initial and Final Pressures.

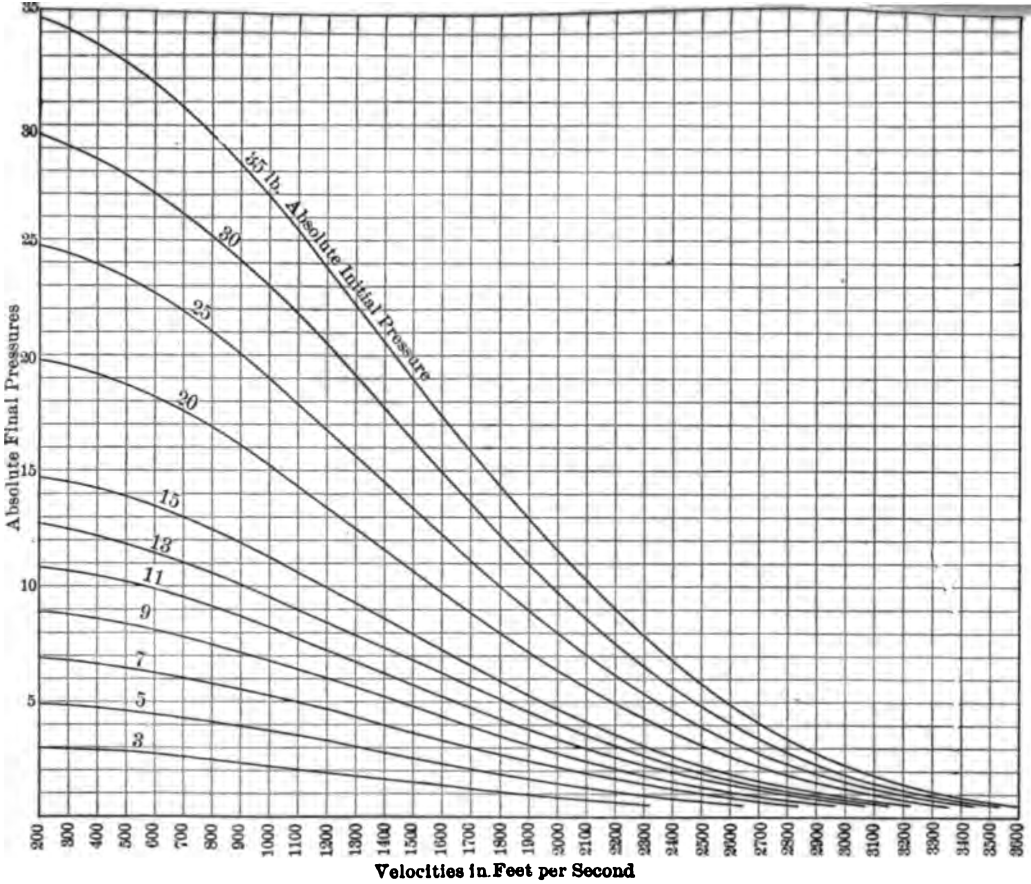


Fig. 4. Velocity of a Steam Jet in Feet per Second

TABLE OF THE PROPERTIES OF SATURATED STEAM.

Pressure, Absolute, Pounds per Square Inch (p).	Temperature, Degrees Fahrenheit (t).	Heat of Liquid (q).	Heat of Vaporization (r)	Total Heat (λ).	Entropy of Water (θ).	Entropy of Steam (ϕ).	Volume of 1 Pound in Cubic Feet (s).	Weight of 1 Cubic Foot in Pounds.
1	2	3	4	5	6	7	8	9
1	101.99	70 0	1043 0	1113 1	0.133	1.985	334.6	0 00299
5	162.34	130 7	1000 8	1131 5	.235	1.842	73.22	01366
10	193 25	161.9	979 0	1140 9	.284	1.781	38.16	.02621
15	213 03	181 8	965 1	1146.9	.314	1.747	26.15	.03826
20	227.95	196 9	954 6	1151.5	.336	1.722	19.91	.05023
25	240 04	209 1	946 0	1155 1	.354	1.704	16.13	.06199
30	250 27	219 4	938 9	1158 3	.369	1.689	13.59	.07360
35	259.19	228 4	932 6	1161 0	.381	1.677	11.75	.08508
40	267.13	236 4	927.0	1163 4	.392	1 666	10.37	.09644
45	274.29	243 6	922 0	1165 6	.402	1 656	9.287	.1077
50	280.85	250 2	917.4	1167 6	.411	1 648	8 414	.1188
55	286.89	256 3	913 1	1169.4	.419	1.640	7 696	.1299
60	292.51	261 9	909 3	1171.2	.427	1.634	7.096	.1409
65	297.77	267 2	905 5	1172.7	.434	1.628	6.583	.1519
70	302.71	272 2	902 1	1174 3	.440	1.622	6.144	.1628
75	307.38	276.9	898 8	1175 7	.446	1 617	5.762	.1736
80	311.80	281 4	895 6	1177 0	.452	1 612	5.425	.1843
85	316.02	285 8	892 5	1178 3	.458	1.607	5.125	1951
90	320 04	290 0	889 6	1179 6	.463	1 603	4.858	.2058
100	327 58	297 9	884 0	1181 9	473	1 595	4 403	.2271
115	337.86	308.7	876 3	1185 0	.487	1 584	3 862	.2589
135	350.03	321.4	867.3	1188 7	.503	1.572	3 323	.3009
140	352.85	324 4	865 1	1189.5	.506	1 570	3 212	.3113
150	358 26	330.0	861 2	1191.2	.513	1 565	3.011	.3221
165	365.88	338 0	855.6	1193.6	.523	1 558	2.751	.3635
175	370 65	343 0	852 0	1195.0	.529	1 554	2.603	.3841
185	375 23	347.2	848 6	1196.6	.535	1 550	2 470	.4049
190	377 44	350 1	847 0	1197 1	.538	1 549	2.408	.4153
200	381 73	354 6	843 8	1198 4	543	1.544	2.294	.4359
215	387.88	361 0	839 2	1200 2	550	1.539	2.142	.4669

Abridged from the tables of Prof. C. H. Peabody, with his permission. Values of the entropy of steam taken, by permission, from *Heat and Heat-engines*, by Prof. F. R. Hutton. Both of the foregoing are published by John Wiley and Sons, New York.

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